AN EXPERIMENTAL SET-UP FOR TESTING CAPILLARIES FOR DIFFERENT REFRIGERANTS

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Certificate

Certified that the work contained in the thesis entitled **"AN EXPERIMENTAL SET-UP FOR TESTING CAPILLARIES FOR DIFFERENT REFRIGERANTS"**, by Major Shalabh Gupta, has been carried out under my supervision and that this work has not been submitted elsewhere for a degree.

Keshav Kant

Dec. 26, 2001

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Dedicated to

My wife and son

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SYNOPSIS

The present thesis work has been conceptualized as the stepping stone for creating an elaborate data base for capillary tubes of different configurations, as expansion device for different refrigerants for a 2 TR refrigeration system. The present work is a part of a research programme aimed in this direction.

The basic aim of the data base is to provide a ready reference to the practicing engineers in the field of Refrigeration and Air-conditioning, as regards the length, diameter and shape of the capillary section to be used as expansion device for systems of 2 TR capacity for different operating conditions and refrigerants.

To create such an elaborate data, extensive experimentation work has to be carried out over a considerable period of time. For this purpose, a sturdy and a robust test rig has to be designed and developed before any meaningful experiments could be carried out.

A test rig consisting of a 2 TR refrigeration system has been designed and developed to carry out the experiments with different configurations of the capillary tubes and create the aforesaid data base. Almost all the devices and equipments required to generate the necessary data have been incorporated to generate the desired data, but the test rig can still be better instrumented.

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Nomenclature OF Various Components

Route 1	-	Flow of refrigerant through thermostatic expansion valve.
Route 2	-	Flow of refrigerant through the test specimen.
V1 to V24	-	Hoke's Needle Valve.
S1 to S4	-	Solenoid Valves.
DV1 to DV9	-	Hand Operated Valves for Control of Refrigerant flow through the system.
'T' junction 'A'	-	Copper 'T' of the 'U' connection at the top of the flow meter.
'T' junction 'B'	-	Copper 'T' of the 'U' connection at the bottom of the flow meter.
'T' junction No.1	-	Copper 'T' at the bifurcation of Route No.1 and Route No. 2 before the expansion device.
'T' junction No.2	-	Copper 'T' at the merging point of Route No.1 and Route No. 2 after the expansion device.
T1 to T21	-	Thermocouples at various points of the main line and the test section.
P1 To P6	-	Pressure tapping at various points on the main line.

Chapter1

Introduction

1.1 Capillary tube as a flow control device

Control of refrigerant flow is an essential feature of all refrigeration systems. All the mechanical refrigerating units have a pressure reducing arrangement or a throttling device which meters the flow of refrigerant to the low pressure side in accordance with the demands placed on the system. This constitutes one of the four basic components of a refrigeration system, viz. the compressor, the condenser, the expansion valve or the throttling device and the evaporator. The thermostatic expansion valve has proved to be a successful throttling device (excluding the case of flooded evaporators), but is quite costly and complicated. However, with the advent of hermetic type compressor and the halocarbon refrigerants, the capillary tube flow control became practicable and popular. This has been used commonly with small units like household refrigerators, food freezers, water coolers, dehumidifiers and room air conditioners, i.e. in applications of less than 10 kW capacities. More recently, it has also been used with larger units, up to 10 TR capacity.

The capillary tube is placed between the condenser and the evaporator as the expansion device to throttle the liquid refrigerant coming from the condenser down to evaporator pressure. The restriction through this tube is sufficient to hold back high pressure in the condenser and, at the same time, it allows liquid to bleed to the evaporator. The passage of liquid refrigerant through the capillary is much easier, than the refrigerant vapor, due to its higher density. Infact, a cubic centimeter of liquid refrigerant weighs 20 to 40 times as compared to that of a cubic centimeter of its vapor depending upon the pressure.

The most efficient operation of the capillary tube occurs only at one particular combination of condenser and evaporator pressures. In a properly sized capillary tube, subcooled liquid enters the tube giving the liquid seal at the entrance. This liquid is forced through the tube due to condenser pressure. There is a pressure drop along the length of the tube. When the pressure drops to a point where the refrigerant becomes saturated, flash gas forms. The formation of flash gas increases as the liquid moves further. This mixture of refrigerant liquid and vapor is delivered to the evaporator at a pressure slightly above that of the evaporator. The part of the capillary tube containing only the liquid refrigerant is called the liquid length. The point at which the first bubble of the flash gas forms is called the bubble point. The remaining part of the tube which contains both liquid and vapor is called the two phase length. All the two phase flow patterns occur in this portion of the capillary tube.

If the condenser pressure increases or the evaporator pressure decreases the flow in the capillary is increased and there occurs a little or no subcooling at all. As a result, the bubble point moves closer to the entrance of the tube, which increases the overall restriction and decreases the flow. Such self-regulating characteristics enable the capillary tube to work so well under a variety of operating conditions.

The most outstanding advantages of a capillary tube are the following:-

(i) It is inexpensive and simple in construction,

(ii) It has no moving parts, and

(iii) It allows high side pressure to unload or balance out the low side pressure during the off-cycle, when used in the system having a hermetically sealed compressor. This permits the compressor to start in an unloaded condition allowing the use of low starting torque motor. The need for a separate unloading device on the compressor which is complicated, costly, and troublesome is, thus, eliminated. It is evident, therefore, that the use of a capillary tube results in considerable saving in the manufacturing cost of a refrigerating system.

The successful operation of a well-designed capillary system is dependent upon:-

(i) Keeping the system fully charged with refrigerant over its working life,

(ii) Maintenance of high standards of internal cleanliness and dehydration,

(iii) The efficacy of the compressor to mix the oil at a uniform rate with the refrigerant.

In a hermetically sealed compressor with a capillary tube, all the above conditions are fulfilled. The same however is not true with an open type compressor, which may lose sufficient refrigerant by leakage past the shaft seals, thereby, requiring occasional field servicing.

The following factors influence the flow of refrigerant through a capillary tube:-

1. Capillary factors

(i) Diameter,

(ii) Length,

(iii) Entrance restrictions,

(iv) Internal finish, roughness, etc.

2. System Factors

(i) Velocity at the capillary entrance,

(ii) Pressure difference across the capillary,

(iii) Condition of refrigerant entering the capillary (sub cooled or saturated),

(iv) Heat flow along the capillary (from outside or from heat exchanger),

(v) Refrigerant properties (density and viscosity),

(vi) Lubricant properties (oil circulation rate, oil viscosity, and miscibility of oil with refrigerant).

It is thus obvious that despite the capillary tube's physical simplicity, it is essential that a refrigeration system employing a capillary tube be carefully designed to obtain satisfactory operation. Not only should the capillary and the heat exchanger assembly be properly matched for capacity balance, but both the high and low pressure sides should also be properly designed for optimal performance.

1.2 Motivation and Scope of the Study

The present thesis work has been conceptualized as the stepping stone for creating an elaborate data base for capillary tubes as expansion device for different refrigerants for a 2 Ton refrigeration system. The present work is a part of a research programme, the ultimate aim of which is to create the data base.

What is this data base? As we all know that the amount of cooling produced in the evaporator depends upon the following factors:-

(i) Compressor capacity,

(ii) Evaporator size,

(iii) Mass flow rate of the refrigerant through the system which would further depend on the capillary tube internal roughness, its length, diameter, shape, etc.

Whereas the compressor size (it's capacity) and the evaporator size could be changed along with other components as per the requirement, the mass flow rate can be varied by changing the different parameters of the capillary tube. Tables can therefore be created for a particular evaporator and condenser pressure giving out mass flow rate of the refrigerant for a particular length, diameter and shape of the capillary tube. This sort of tables can be prepared for different condenser and evaporator pressures catering for all seasons of the year. It can easily be imagined that this data base shall serve as a ready reference for all practicing Air-conditioning and Refrigeration Engineers as regards the length, diameter and shape of the capillary tube for a particular mass flow rate required. Another factor which can subsequently be included in the data base is the evaporator temperature, (i.e. the temperature that can be maintained) by varying the capillary parameters.

The literature search on the subject shows that not much data are available for ready reference for use of a capillary as an expansion device for all the refrigerants in use today in refrigeration industry except for R-12 and R-134(a) to a limited extent. The idea of creating a data base for capillary as regards its length, diameter and shape was conceived as a part of the ongoing research in the area of new refrigerants at Indian Institute of Technology, Kanpur. These data could be extremely useful to the practising Air-conditioning and Refrigeration Engineers.

With the environmentalist creating a hue and cry over the use of hydrocarbon refrigerants in refrigeration industry, it is now becoming more important to switch over to refrigerants which are more environment friendly. Thus, a need was felt that if a data base could be created for more commonly used refrigerants as regards the use of capillary as expansion device for smaller refrigeration systems, it shall be extremely useful for retrofitting small refrigeration systems with new or alternate refrigerants. Certain amount of data are already available for few of the refrigerants like

R-12 and R-134, there is a need to generate data for few additional refrigerants which are going to be used on much larger scale in very near future. With this in mind, the first requirement was to design and develop an experimental test rig to be able to conduct experiments to first validate the available data with R-12 and R-134 (a) and later to collect more data with other refrigerants. The aim of the thesis was to set up an experimental test rig for testing capillaries for different refrigerants.

It is for this complexity of NACT that only a limited amount of research has been undertaken on these tubes. Whatever work has been reported so far in literature, does not take into account most of the complexities of the flow. Schematic of capillary tube- suction line heat exchanger is shown in figure 2.1.

4. ACT are placed after the condenser as a normal expansion device and do not act as a heat exchanger unlike NACT. The refrigerant is assumed to expand from high pressure to low pressure adiabatically. They are comparatively simple in design and the flow phenomena do not have many complexities (4). Due to the simplicity of the flow pattern, this relatively speaking has been explored much more than NACT. A schematic of vapor compression refrigeration system employing adiabatic capillary tubes is shown in figure 2.2.

5. The present thesis work also aims at developing an experimental set-up for creating a data base for adiabatic capillary tubes and hence in this review some of the experimental and theoretical work reported on ACT only has been discussed briefly, as given in the following sections.

2.2 Experimental Studies on Adiabatic Capillary Tubes

Lathrop (1) presented a capillary tube parametric study based on the compilation of experimental results on refrigerating equipment at Philo Corporation. The paper dealt with a straightforward approach to capillary tube design and provided information for a more accurate solution of problems pertaining to capillary restrictor tubes. The data were presented in a graphical form to depict relations between different parameters like flow rate, degree of sub-cooling of refrigerant, diameter and length of capillary tubes, and pressure differences at inlet and outlet of capillary tubes. He has presented rough empirical relationships. The author also attempted the problem of matching the tube characteristics with the compressor characteristics.

Staebler (2) presented his work on capillary tube selection centered on system balance. Based on his experimental results, he provided few charts and tables to help selection of a capillary tube for use with R-12 and R-22. These charts coupling compressor

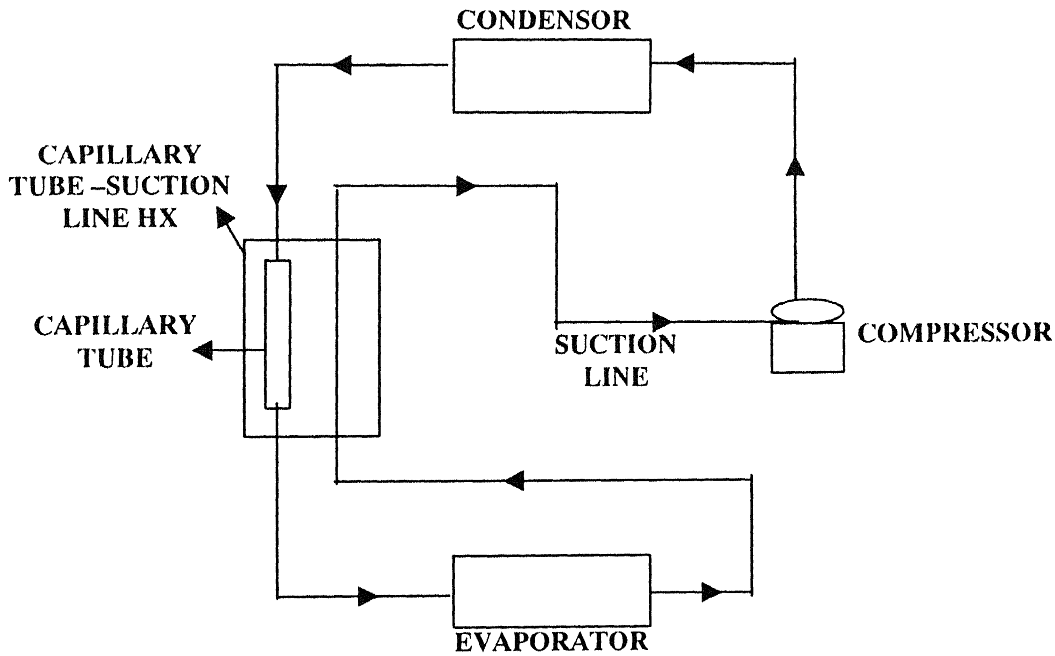


Figure 2.1 Schematic of NACT in a VCRS

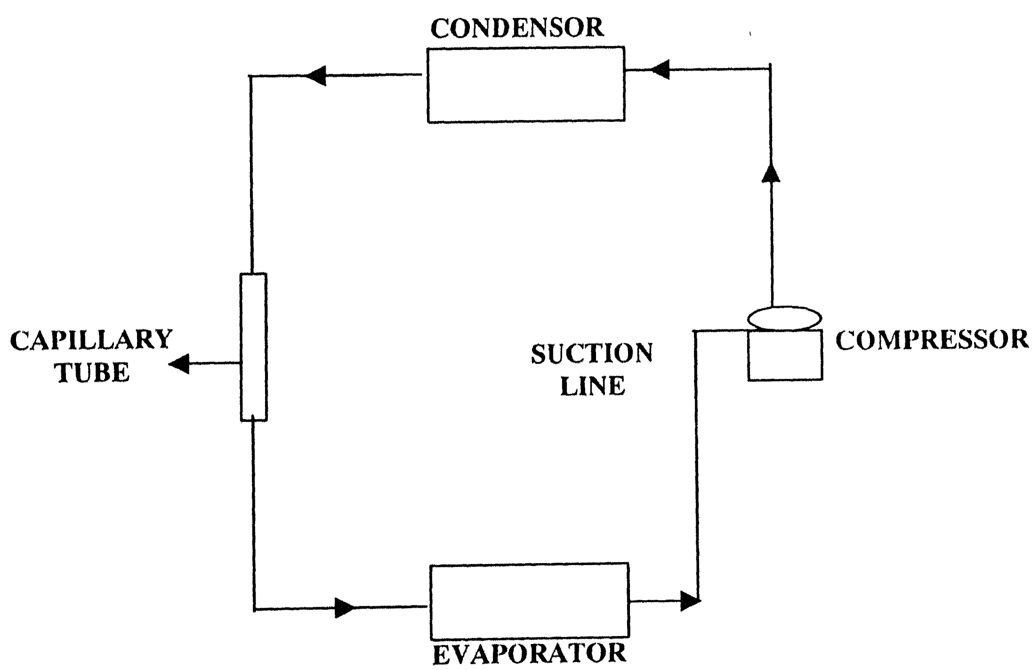


Figure 2.2 Schematic of ACT in a VCRS

capacity to capillary resistance were used to reduce the preliminary testing work necessary in the proper design of a capillary tube for a refrigerating system.

Bailey et al (3) conducted experiments for adiabatic flow through orifices, nozzles and short tubes. They observed the nature of flow of saturated and nearly saturated water into regions where the pressure was less than the saturation pressure. They presented a method of computing the mass flow rate based on the rate of evaporation from a metastable liquid core into a surrounding ring of vapour. They could accurately predict the flow rates for orifices, but the same for the nozzle could be predicted within an error of 10% of the measured value.

Cooper et al (4) performed extensive experimentation and studied capillary tubes of 304.8mm, 457.18mm, 609.5mm and 914.36mm(12 in, 18 in, 24in, and 36in) lengths and internal diameters of .9143553mm and 2.54mm(0.036in and 0.1in). These capillary tubes were tested for four different inlet conditions of inlet pressures of 10.4 bars, 13.5 bars, 16.55 bars, and 19.655 bars (151, 196, 240, and 285 psia) and correspondingly 297.45, 307.45, 315.78, and 323 degree Kelvin (76, 94, 109, and 122 °F) inlet temperatures. They also conducted flow visualization tests to observe the type of two-phase flow. Using naked eye observation, they described the capillary tube flow in two-phase conditions as 'fog flow'. They did not observe the 'slug-flow' or 'bubble flow'.

Kuehl and Goldschmidt (5) studied experimentally several capillary length and diameter combinations using a test loop that provided them control of subcooling, and condenser and evaporator pressures. Their study was focused on adiabatic capillary tubes using R-22, and was directed towards detailed data that would lead to means of selecting capillary tube geometry for the given inlet pressure and temperature (degree of subcooling), flow rate, and evaporator pressure. The pressure distribution was measured. They also made measurements to determine the effects of coiling on the restrictor characteristics of the capillary tube. Coiling was noted to increase the pressure drop by about 5 %, while the loss factor for the distributor was found to be slightly higher than that for re-entrant flows.

Kuehl and Goldschmidt (6) conducted experiments to get the transient response of five different capillary geometries and one orifice using R-22. They subjected them to a step change in operation to determine their transient response characteristics. The time required to achieve steady state was approximately three to five minutes for capillary tube and one minute for the plug orifice tested. The capillary response was found comparable to the total system response time after compressor start-up. The orifice, which demonstrated a response time of approximately 30 to 60 seconds, clearly was superior in terms of time constant for transient operation. The effects on the overall system performance were not considered in the paper.

Li et al (7) investigated the metastable flow phenomenon of R-12 through two capillary tubes 0.66 mm and 1.17 mm internal diameters and 1.5 m length experimentally. They made the following conclusions from the experimental observations:

(1) The larger the diameter of the capillary tube, lower the under pressure of vaporization, and shorter the length of the metastable flow in the single-phase liquid.

(2) The larger the refrigerant mass flux, the larger the under pressure of vaporization.

(3) An increase of the inlet subcooling decreases the under pressure of vaporization.

(4) The location of the inception of vaporization does not change distinctly with the change of the backpressure. The effect of the change of the backpressure on the under pressure of vaporization is small.

(5) For a two-phase flow with vaporization, under the critical flow condition, a drop in the backpressure has an effect on the pressure distribution downstream of the location of the inception of vaporization. This phenomenon is different from that in a single-phase flow.

Melo et al (8) conducted experiments to collect extensive data for flow of CFC-12 and HFC-134a through an adiabatic capillary tube. Capillary tubes of 0.7, 0.8 and 1.0 mm internal diameter with lengths of 3 m and 2 m were tested. Condensing pressures of 18, 14, and 9 bars, and subcooling ranging from 14 °C to 2 °C were chosen as the operating condition. They observed that the tube length, condensing pressure, internal diameter

and subcooling, affected the mass flow rates of both refrigerants. The effect of the length and internal diameter on the mass flow rate seemed to be stronger for HFC-134a than for CFC-12. Both fluids generated almost the same flow rates at the same condensing pressure and subcooling. This behavior was slightly affected by the capillary tube geometry and by the operating conditions.

Melo et al (9) conducted experiments to measure the pressure and temperature profiles for flow of R-134a through adiabatic capillary tube at different condensing pressures and different levels of subcooling. The experimental data were then used to evaluate the suitability of some equations previously reported in literature for the single phase friction factor, the under-pressure of vaporization, and the entrance contraction factor. Correlations for the average and local two-phase friction factors were also developed on the basis of experimental data. Like Lin et al (26) they also confirmed that Churchill's equation was appropriate for the determination of frictional pressure drop coefficient in single-phase flow through adiabatic capillary tubes. In their opinion the equation proposed by Chen et al (9) seemed to be inappropriate for the calculation of underpressure of vaporization in capillary tubes. This was so because the length of non-equilibrium region in a given capillary tube could vary from test to test even if the operating conditions were unchanged. Their calculated entrance contraction loss factor was above the values recommended by the literature for reentrant contractions. The pressure drop associated with a sudden contraction from a 4.75 mm cross-sectional diameter to the capillary tube diameter had a negligible influence on capillary tube flow.

2.3 Numerical studies on adiabatic capillary tubes

Marcy (10) suggested a quite different method for theoretical analysis of flow through adiabatic capillary tube. As his analysis was based on the graphical integration of Fanning equation and, therefore, the solution did not depend on empirical factors. Unlike most of previous researchers, this work did not take care of capacity balance and effect of oil entrainment on the rate of flow through capillary.

Hopkins (11) studied in detail the flow ratings of capillary tube expansion devices for R-12 and R-22. The ratings provided were the results of a study based on the analysis of fluid flow data in a manner similar in several respects to that of Bolstad(29) and Marcy(10) for refrigerants and Benjamin for steam. Using these ratings, one could obtain approximate diameter and length of capillary tube for a particular set of conditions.

Goldstein (12) presented a method to predict the flow characteristics of a small diameter tube undergoing flashing flow due to heat transfer and rapid depressurization. Assuming homogeneous flow through the capillary tube and no vapour blow-by, he developed a model and presented a flow chart which could be used in the modeling of the refrigeration system. Author claimed that for any particular application, the method was sufficiently general to permit adjustments in any factor found to be imprecise by test. System analysis of commercial and residential refrigeration units had shown excellent correlation with manufacturer's data.

Li et al (13) developed a thermodynamic non-equilibrium drift flux model to determine the performance of a steady-state two-phase flow of refrigerant through capillary tube expansion device considering the relative velocity between the liquid and vapour phases. The five basic differential equations based on the drift flux model of two-phase flows were solved simultaneously by using Runge-Kutta method. The comparison of their model with the experimental data available in the literature for the capillary tube flow using R-12 showed that the model was in good agreement with the experimental data and can be used for the selection of capillary tube expansion device used in refrigeration systems.

Escanes et al (14) presented a collection of charts prepared for a preliminary selection of capillary tube expansion device in refrigeration units using R-12, R-22, and R-134a as working fluids. The charts were based on their numerical model assuming one-dimensional steady flow through adiabatic capillary tubes. They used basic conservation equations and step by step numerical scheme. For a given set of input data; working fluid, capillary length and inside diameter and wall roughness, degree of

subcooling, condenser and evaporator pressures, the model could predict flow condition (choked or not), flow variables along the tube (temperature, pressure, velocity, dryness fraction, void fraction etc.), mass flow rate, and refrigerating power. The predictions of the model were compared with the experimental data available in literature for R-12 and R-22, which agreed well.

Peixoto et al (15) presented a finite difference model for capillary tube-suction line heat exchangers that could be used for either simulation or design. Model was validated using the available data for R-12 and found to be in good agreement. The experimental data available were for a high degree of subcooling resulting in a large liquid region; the model therefore predicted the liquid flow perfectly. As claimed by the authors the model was valid for alternative refrigerants as well. For a specific refrigerant, the model permitted the choice of three dependant variables. For a given set of operational and geometric conditions, the model calculated mass flow rate, capillary tube exit quality and suction line temperature at heat exchanger outlet. For design applications, the model could be used to determine the tube diameter and length required for the given operational conditions and mass flow rate. With the help of plots they compared the performance of lateral tube and concentric tube type of heat exchangers for R-12 and R-134a.

Wong et al (16) predicted the effects of various design parameters on the capillary tube. Assuming homogeneous two-phase flow through adiabatic capillary tube, a theoretical model was developed. The model was used to illustrate various effects of the design parameters such as: capillary diameter, length, and its wall roughness, degree of sub cooling, and mass flow of refrigerant on the capillary tube performance. The predicted pressure distribution and critical mass flow were compared with the measured values available in literature. The variation of various parameters mentioned above along the capillary length was also presented.

Escanes et al (17) developed a numerical model to simulate the thermal and fluid-dynamic behavior of capillary tube expansion device. The basic governing equations of the flow (continuity, momentum and energy) were written in one dimensional and

transient form across the control volumes and solved using an implicit step-by-step numerical technique. The formulation required the use of additional equations to evaluate the convective heat transfer, shear stress, and void fraction. A special treatment of the point of transition between single-phase and two-phase fluid flow was implemented together with a Newton-Raphson algorithm for the evaluation of the inlet mass flow rate corresponding to the given boundary conditions. Their model allowed analysis of aspects such as capillary geometry, type of fluid, critical and non-critical flow conditions, boundary conditions (heat transfer, condenser and evaporator pressures etc.) and transient aspects. Model was in good agreement with the experimental data of three authors (Bolstand and Jordan, Whitesal, and Mikol).

Wong et al (18) presented a homogeneous model using various correlations for the determination of two-phase viscosity. They were of the opinion that the selection of an appropriate two-phase friction factor with the use of a suitable two-phase viscosity correlation was important to effectively apply the homogeneous models. In the process, they mentioned the findings of various researchers who worked on capillary tube expansion with the assumption of homogeneous two-phase flow. The viscosity correlations given by Owen, McAdam, Dukler, Li et al, Cicchitti, and Beatti & Whalley were used in the model and the predicted pressure drops along the capillary tube were compared with the experimental data available in the literature. It was found that predictions were having minimum RMS error when the viscosity correlation by Duckler et al was used in the model.

Bansal et al (19) developed an empirical model to size adiabatic and non-adiabatic capillary tube expansion devices for small capacity vapour compression refrigeration systems. The model was based on the assumption that the length of capillary tube was a function of five primary variables, namely the inner diameter, inner wall roughness, mass flow rate, degree of subcooling and the pressure difference between condenser and evaporator. The model was validated with the experimental data and theoretical predictions available in the literature for the refrigerant HFC-134a and found to agree reasonably well (maximum variation with the previous works was ± 8%.

Wong et al (20) did a theoretical study for flow through an adiabatic capillary tube assuming it to be homogeneous. In this study, they compared the flow characteristics of R-12 with that of R-134a. The results revealed that even with the minor differences in their refrigerant properties, the differences in their flow characteristics were significant. They found that R-134a had higher pressure drop than R-12 both in the single-phase and two-phase flow regions and therefore for the same operating conditions, R-134a would require smaller capillary length (about 15 to 20% reduction was expected). It was also predicted that for the same tube dimensions and conditions, R-12 could accommodate a higher mass flow rate before reaching the choked flow condition. Thus for the same critical tube length, the mass flow rate of R-12 was estimated to be at least 15% higher than that of R-134a.

Wong et al (21) presented a theoretical comparison of homogeneous flow and separated flow models using the experimental data available in the literature. They observed that the Moody's friction factor was applicable for predicting the pressure drop for the single-phase flow region of the small diameter capillary tube. For the two-phase flow through capillary tube, they showed that the homogeneous model and the separated flow models both might be used to predict the various parameters. They concluded that the separated flow model using Miropolskiy's slip ratio and Lin's correlation for pressure gradient gave better predictions than the homogeneous model. It was observed that greater discrepancies occurred in the region near the choked condition while using the homogeneous model.

Bansal et al (22) presented a homogeneous two-phase flow model CAPIL to study the performance of adiabatic capillary tube expansion device used in small refrigeration systems. The model was based on one-dimensional conservation equations of mass, momentum and energy that were solved simultaneously through an iterative method and Simpson's rule. The model used empirical correlations for single-phase and two-phase friction factors and gave due consideration to pressure loss due to sudden contraction at the entrance of the capillary tube. The model used the REFPROP database for refrigerant properties. The design parameters for flow through capillary like tube length, tube diameter, relative roughness, degree of sub cooling, and

refrigerant mass flow rate were considered. The model was validated with earlier models over a wide range of operating conditions and was found to agree reasonably well with the experimental data for R-134a available in the literature (within ± 10%).

Sami et al (23) presented a numerical model for predicting the capillary tube performance using R-12, R-22 as well as new alternative refrigerants, namely pure and/or binary mixtures. The model was established after the fluid flow conservation equations written for a homogeneous refrigerant flow under sub cooled, saturated, and two-phase fluid conditions. The predictions from the model for the investigated conditions agreed well with the available experimental data taken from literature.

Chen et al (24) developed rating correlations for R-134a flowing through adiabatic capillary tubes. The ratings were based on the curve fittings of capillary theoretical predictions and considered both the geometric and system operating factors. The proposed correlation in the form of algebraic equations, were quite simple and versatile. They could be used to calculate the mass flux and pressure drop directly for a given capillary tube. On the other hand, optimum capillary size corresponding to any given set of operating conditions could also be calculated. The correlations were validated using two sets of experimental data, and their predictions agreed well with the measured values. They explained the method of using the correlations to rate the capillary tube expansion device.

Jung et al (25) modeled the pressure drop through a capillary tube to attempt the prediction of the size of the capillary and to provide simple correlations for practicing engineers. Stoecker's model was modified with the consideration of various effects due to subcooling, sudden contraction of area at capillary inlet, different equations for viscosity and friction factor, and finally mixture effects. McAdams' equation for the two-phase viscosity and Stoecker's equation for the friction factor yielded the best results among various equations considered for two-phase viscosity and friction factor. The model yielded the performance data that were comparable to those in the ASHRAE handbook. After the model was validated with experimental data for R-12, R-134a, R-22, and R-407c, performance data were generated for R-22 and its alternatives, R-134a,

R-407c, and R-410A under the following operating conditions: condensing temperature of 40, 45, 50, and 55°C; sub cooling of 0, 2.5, and 5°C; capillary tube diameter from 1.2-2.4 mm, and mass flow rate of 5-50 g/s. These data showed that the capillary tube length varied uniformly with the changes in the condensing temperature and subcooling. A regression analysis was performed to determine the dependence of the mass flow rate on the length and diameter of a capillary tube, condensing temperature, and degree of subcooling. They determined simple practical equations, yielded a mean deviation of 2.4 % for 1488 data obtained for two pure and two mixed refrigerants examined in the study.

Wongwises et al (26) developed a two-phase separated flow model to determine the refrigerant flow characteristics in adiabatic capillary tubes. Starting from basic conservation equations partial differential equations were derived and solved simultaneously using fourth order Runge-Kutta method. The simulated results were compared with the experimental data available for capillary tube flow with R-12, R-22 and R-134a as working fluid. It was observed that the separated flow model with appropriate correlations of frictional pressure gradient and slip ratio could be satisfactorily used to predict the two-phase flow for adiabatic capillary tube expansion devices.

Wongwises et al (27) starting with the basic conservation equations, theoretically compared the flow characteristics of several pairs of refrigerants flowing through adiabatic capillary tubes assuming a homogeneous two-phase flow. Two-phase friction factor was determined using Colebrook correlation. The partial differential equations were simultaneously solved using the fourth order Runge-Kutta method. By varying the model input parameters for all pairs, it was found that the traditional refrigerants consistently gave lower pressure drops for both single-phase and two-phase regions, which resulted in longer capillary lengths.

Liang et al (28) presented an equilibrium two-phase drift flux model to simulate the flow of refrigerant through adiabatic capillary tubes. They predicted the parameters for R-12 and R-134a and compared them with the experimental values available in the

literature for R-12. Variation of flow characteristics like pressure, void fraction, dryness fraction, velocities of liquid and vapor phases, and their drift velocities along the length of capillary tube was presented for R-134a.

2.4 Theoretical and Experimental studies on Adiabatic capillary tubes

Bolstad and Jordan (29) performed experiments on adiabatic capillary tubes. Their work may be treated as first major study on capillary tube flow. Capillary tubes of five different diameters varying from .66mm to 1.397mm (0.026 in to 0.055 in), and lengths of 1.829m, 3.657m and 5.487m (6,12, and 18 feet) were tested. The working inlet pressures of 8.27 bars, 9.65517 bars and 11.03448 bars (120,140, and 160 psia) were used with R-12 as the working fluid. They observed that for adiabatic flow for a given inlet pressure and temperature, there existed a critical pressure at the exit (just before the exit) and the pressure cannot be decreased beyond this critical value. If the exit pressure was equal to the critical value, the mass flow rate was maximum. Any decrease in the evaporator/exit pressure did not affect the flow rate though the tube. It was observed that the flow rate was dependant on inlet conditions, exit pressure and the oil content of the refrigerant. The presence of oil in the liquid refrigerant increased the flow rate.

The authors also attempted the theoretical analysis of fluid flow through capillary tube to have better understanding of physical phenomenon of refrigerant flow through the capillary tube as the expansion device. Assuming one-dimensional flow through horizontal tubes, they suggested the following equation to determine the capillary tube length.

$$dl = (Tds - dq)\, J / (fv^2 / 2gd)\, (m^*/A)^2$$

As they could not integrate the relation to get total length, they used a graphical method in which they plotted the factor $(2gd/f)\,(A/m^*)^2\,(T/v^2)$ versus entropy. The area below the plot was taken as the total length of capillary tube for the used set of data.

Whitesel (30) studied simultaneous flow of gas and liquid through capillaries and developed two general formulae analytically for slug flow conditions. The first formula was derived for flow of a non-condensable gas and a non-volatile liquid while the

second was developed on the same lines for R-12. The authors carried out an experimental investigation using air and water for the first formula and R-12 for the second. The R-12 formula according to the author is applicable at all inlet qualities, all inlet and outlet pressures, all capillary dimensions and in the temperature range 233 to 333 degree Kelvin (–40 °F to 140 °F). The authors also obtained a secondary empirical formula for computing the two-phase flow friction factors for R-12. The formula is as given below:

$$f_{TP} = 1.5\ (\ f_l(1-x) + f_g\ x)$$

Where, f_l and f_g are the values of Fanning friction factors for liquid and gas respectively, evaluated at the average velocity of mixture at the capillary entrance with the assumption of unit slip ratio. A multiplier of 1.5 gave the increase in the friction factor due to two-phase flow over single-phase value.

Whitesel (31) developed a two-phase flow friction factor formula for R-22 in a more or less same way as that for R-12 in his first paper. He conducted experiments to cover wider range of capillary tube flow parameters and used these test data to modify his earlier two-phase friction factor formula. He gave relation for determination of critical exit pressure for maximum mass flow rate through capillary tube. He provided tables and charts to facilitate the calculations. Unlike previous workers, the work of Whitesel is unique in the sense that his theoretical predictions did not require step-wise integration.

Mikol (32) presented results of his experimental and theoretical work on adiabatic capillary tube. He studied single and two-phase flow for R-12 as well as R-22. He also conducted visual and photographic studies of the two-phase flow phenomenon of R-12 using a glass tube of 1.24mm (0.049in ID) and 1.829m (6 ft) length. He conducted experiments on 1.41mm (0.055509in) ID drawn copper tube of 1.829m (6 ft) length. The results led to the following conclusions:

(1) Drawn copper tubes of small diameters, in general, could not be considered smooth for friction factor purposes.

(2) Fluid flow through small diameter tubes confirmed to continuum flow equations as established for large bore tubes and pipes.

(3) Friction factor correlation of Moody and of others consistent with Moody's correlation was applicable to single-phase flow in small diameter tubes.

(4) The phenomenon of metastability, which was described as the persistence of the liquid state at pressures less than the saturation pressures corresponding to their temperatures, was observed in all test runs and during all visual flow observations. Therefore this phenomenon must be considered in refrigerant capillary design.

(5) The flow pattern in the capillary could be described as a stable mode of sub-cooled liquid flow, metastable liquid flow, a flash point, that starts with vapour bubbles appearing at the tube wall and merging into a vapour core surrounded by liquid, which finally grows into a homogeneous vapour mist near the tube exit.

The author recommended that the refrigerant capillary should be designed according to the stable mode.

Author suggested a tentative correlation for apparent friction factor for two-phase flow (as given below)

$F_{TP(apparent)} / f_l \cong 0.95$

This value agreed within ± 10% with the integrated average two-phase friction factor value computed from the test data.

The author also discussed the choking phenomenon in two-phase flow and he was of the opinion that similar phenomenon occurred due to the same reason as in the gaseous flow. As further expansion took place beyond the capillary exit, the velocity was higher than the sonic velocity at capillary exit. The curves (based on the following formula) for getting the speed of sound in homogeneous two-phase mixture of R-12 were also given in the paper.

$$c = (g_c \, (dp/d\rho)_{s=constant})^{0.5}$$

Koizumi et al (33) performed experiments on adiabatic capillary tubes using R-22 as the working fluid. They made the flow visualization studies by using a glass-capillary

tube and reconfirmed that the flow was homogeneous. They measured the pressure and temperature distribution along the capillary tube and observed that the starting point of vaporization was delayed. They found that a theoretically calculated pressure distribution, presupposing adiabatic homogeneous flow, agreed accurately with measured values, if the delay of vaporization was estimated correctly. Regarding delay of vaporization, they pointed out the followings:

(1) The distance along the tube from the theoretical vaporization point to the practical point (superheated liquid length) decreases and the difference between the superheated liquid temperature and the saturation temperature for the superheated liquid pressure (delay of vaporization) increases with increasing refrigerant flow rate.

(2) The superheated liquid length decreases and the delay of vaporization increases with decreasing inside diameter of the tube under the condition of a same refrigerant flow rate.

(3) For their experiment the delay of vaporization was from 2 °C to 4 °C and the maximum superheated liquid length of 60 cm was observed.

They then suggested a simple method to calculate the length of capillary tube considering the phenomenon of the delay of vaporization and observed that the predicted values are in good agreement with the experiment.

Chen et al (34) investigated the metastable flow phenomenon of R-12 through capillary tubes experimentally. Two capillary tubes of 0.66 mm (0.026 in), and 1.17 mm (0.046 in) internal diameter and 1.5m (4.92 ft) length were used in the experimentation. The relative error as reported by them was 26%. They developed a method to predict the heterogeneous nucleation of refrigerant flow using the classic nucleation theory. They suggested a correlation for the determination of the pressure difference between the thermodynamic saturation point and the point of inception of vaporization.

Lin et al (35) investigated experimentally the local frictional pressure drop during vaporization of R-12 along an adiabatic capillary tube expansion device. They also presented a model for the prediction of the local frictional pressure drop of two-phase flow through capillary tube; the results obtained from the model were in good

agreement with the experimental data with a standard relative error of about 15%. They observed that the variation of the quality along the capillary tube was non-linear; the quality increased more rapidly close to the exit of the tube. For single-phase liquid flow through capillary, they observed that the roughness of the wall surface (inside) had a distinct effect on the frictional pressure drop coefficient. Their experimental results confirmed that Churchill's equation (41) for the determination of frictional pressure drop coefficient to single-phase flow through adiabatic capillary tubes.

Dirik et al (36) did experimental and numerical studies on adiabatic and non-adiabatic capillary tubes using HFC-134a as working fluid. Their study focussed on the effects of suction line heat exchange on flow characteristics of capillary tubes for simulating real operating conditions of household refrigerators. The capillary tube-suction line heat exchanger assembly considered in this study was a double-pipe arrangement in which a portion of capillary tube passes inside the suction line to form counter flow heat exchanger. The flow rates predicted by their numerical model showed good agreement with their experimental data and with the then recent adiabatic measurements reported in literature.

Paiva et al (37) did experimental and numerical studies for the non-adiabatic capillary tube using CFC-12 and HFC-134a. They used lateral and concentric capillary tube-suction line heat exchangers. The effect of inlet subcooling was analyzed and as expected mass flow rate increased as subcooling was increased. When CFC-12 was substituted by HFC-134a, numerical results showed a very slight decrease in the mass flow rate for the lateral heat exchanger and a slight increase for the concentric-tube heat exchanger.

Chang et al (38) investigated the pressure drop for flow through a capillary tube experimentally. Pure refrigerants such as R-32, R-125, and R-134a and their mixtures such as R-32/R-134a (30/70 by mass fraction), R-32/R-125 (60/40), R-125/R-134a (30/70), and R-32/R/125/R-134a (23/25/52) were used as test fluids. The range of Reynolds number of refrigerant flow was between 2×10^4 and 2×10^5. In this range, friction factor was dependent, both on the Reynolds number and the wall roughness.

The binary interaction parameters for viscosities of the liquid state of refrigerant mixtures were found based upon the experimental data in the open literature. They pointed out that for high liquid flow range such as in their experimental study, the homogeneous flow model was appropriate. Several homogeneous flow models predicting the viscosity of two-phase region were compared and Cicchitti's equation was shown to be the most adequate for adiabatic capillary tube flow. A model for the prediction of the frictional pressure drop of single and two-phase flow was presented for the refrigerant mixtures. It was observed that the pressure drop increased as saturation pressure of refrigerant increased while keeping condensation temperature unchanged. The pressure drop for R-32 along the capillary tube was the largest and that for the R-134a was found to be the smallest.

Melo et al (39) performed the experiments on adiabatic capillary tubes using three refrigerants, namely CFC-12, HFC-134a, and HC-600a, and at different condensing pressures and levels of sub cooling under choked flow conditions. The investigations included the effect of capillary tube length and diameter, refrigerant subcooling, condensing pressures and type of refrigerant on mass flow-rates through the capillary tubes. Eight capillary tubes with different combinations of lengths, diameters and tube roughness were used and extensive data (exceeding 1000 sets) were reported. They performed a conventional dimensional analysis to derive correlations to predict the mass flow rates for different refrigerants at different operating conditions. The predictions from the developed correlations were found to be in good agreement with the measured data and other studies in literature. The correlations seemed to be very valuable in the design of adiabatic capillary tubes for alternative refrigerants in future.

Wei et al (40) made experimental study to examine the performance of capillary tubes using R-407 as working fluid. They proposed a correlation with a mean deviation of 2.78 % to correlate the experimental data. Their work was one among very few to look at the effects of coiling on the performance of a capillary tube. A correlation was proposed to describe the relation between the mass flow rates through the straight and coiled capillary tubes.

2.5 Conclusions

For the reasons mentioned earlier an extensive amount of work has been reported on adiabatic capillary tubes for refrigerants like R-12, R-22 and R-134. Due to the environmental problems posed by these CFC refrigerants, there is a need to take up both experimental and theoretical research with newer refrigerants on an urgent basis and come out with some meaningful recommendations for the existing VCRS.

Chapter 3

Experimental Set-Up

3.1 General

The experimental set-up comprises, basically of a 2-ton capacity vapour-compression refrigeration system. Added on to the basic structure of the refrigeration system comprising of a compressor, condenser, expansion device (capillary- referred to as test section) and an evaporator, are the various instruments designed to make measurements. The set-up is flexible enough to enable the testing of different types and sizes of the test specimen.

A schematic diagram of the set-up is shown in figure 3.1. Photographic view of the experimental set-up is presented in figures 3.2 to 3.7. The complete set-up with different components has been installed on a basic framework of 76.2 mm x 76.2 mm (3"x3") angle iron as can be seen from the photographs of the set-up.

3.2 Working

Mixture of the oil and refrigerant (R-134 in this case), delivered from the compressor is allowed to pass through the oil separator where the oil associated with the refrigerant is separated. The pure refrigerant vapour is then allowed to enter the water cooled condenser where it losses heat to the cooling water and reduces to the state of saturated liquid. The condensed liquid refrigerant is then stored in the receiver and passed through the flowmeter (incorporated to measure the mass flow rate of the refrigerant in the system). The liquid refrigerant is then passed through the drier to remove any moisture content that may be present in the refrigerant and may cause choking of the test specimen. The refrigerant at this stage is still liquid which is now passed through the expansion device (two routes-one through the thermostatic expansion valve and the other through the test specimen). For test running of the system, i.e., to find leakages etc, Route 1 is taken and while doing actual performance

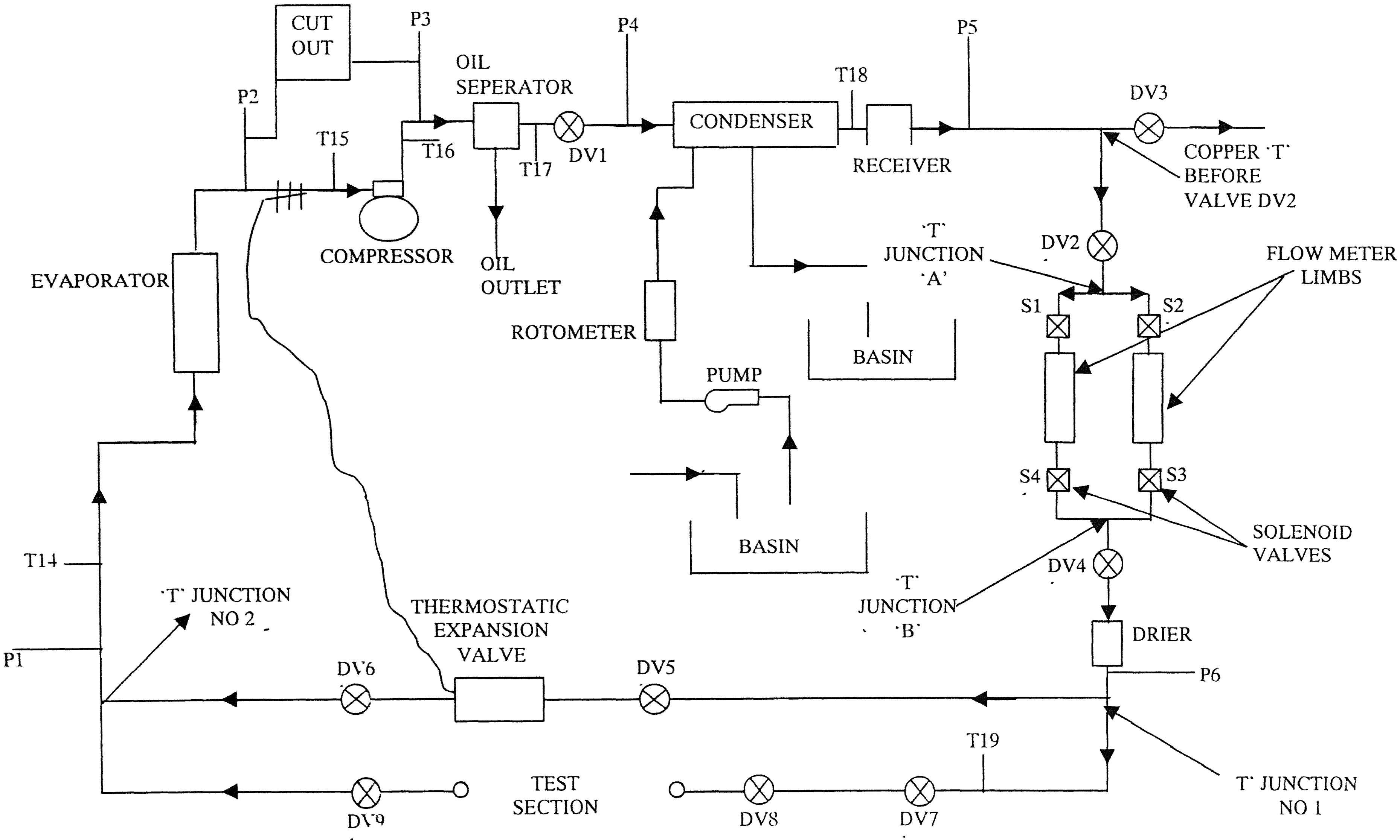


Figure 3.1 Schematic Layout of the Experimental Setup

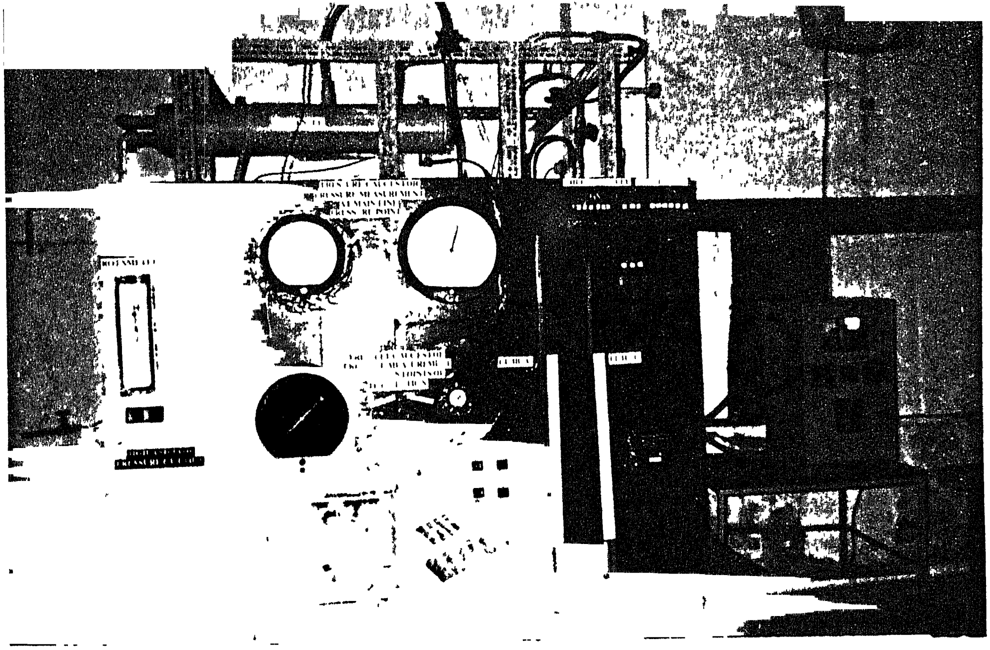


Figure 3.2. Front view photograph of the system without insulation

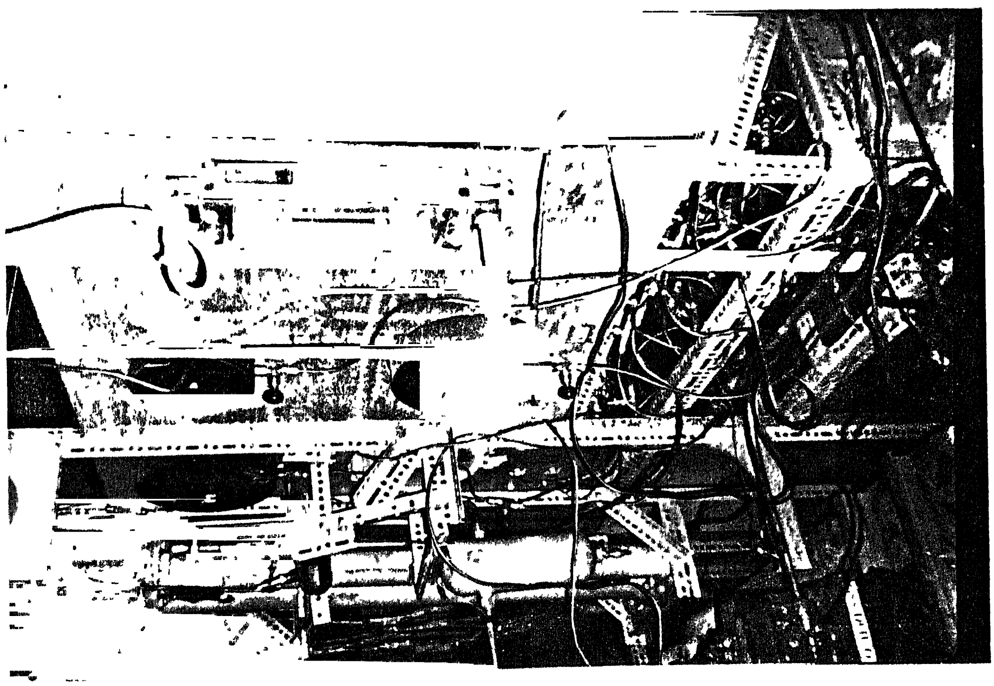


Figure 3.3. Rear view photograph of the system without insulation

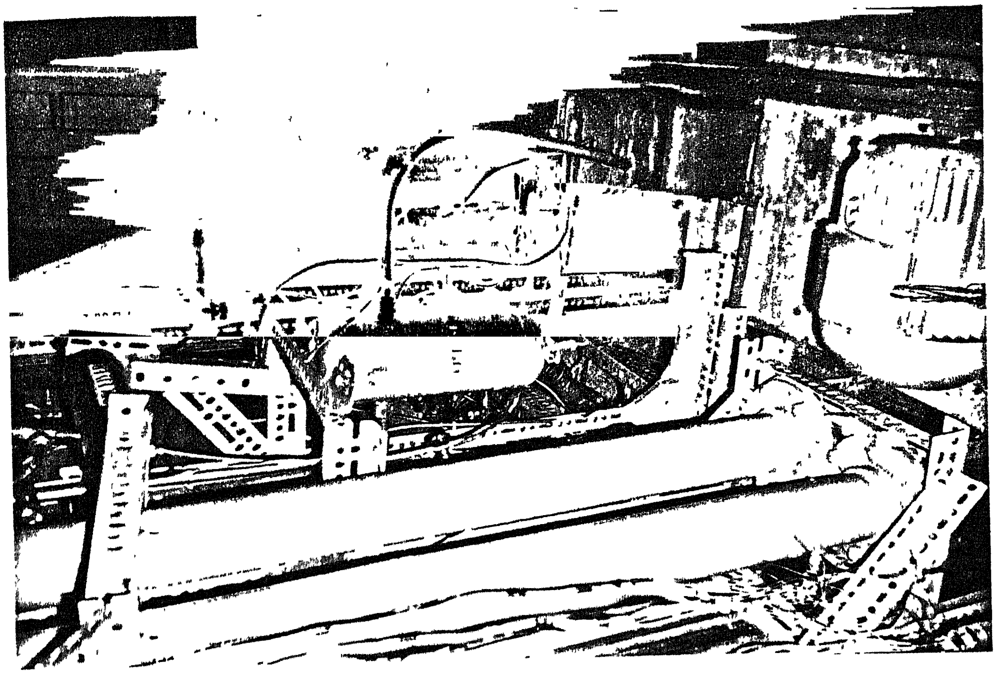


Figure 3.4. Rear view photograph of the system without insulation

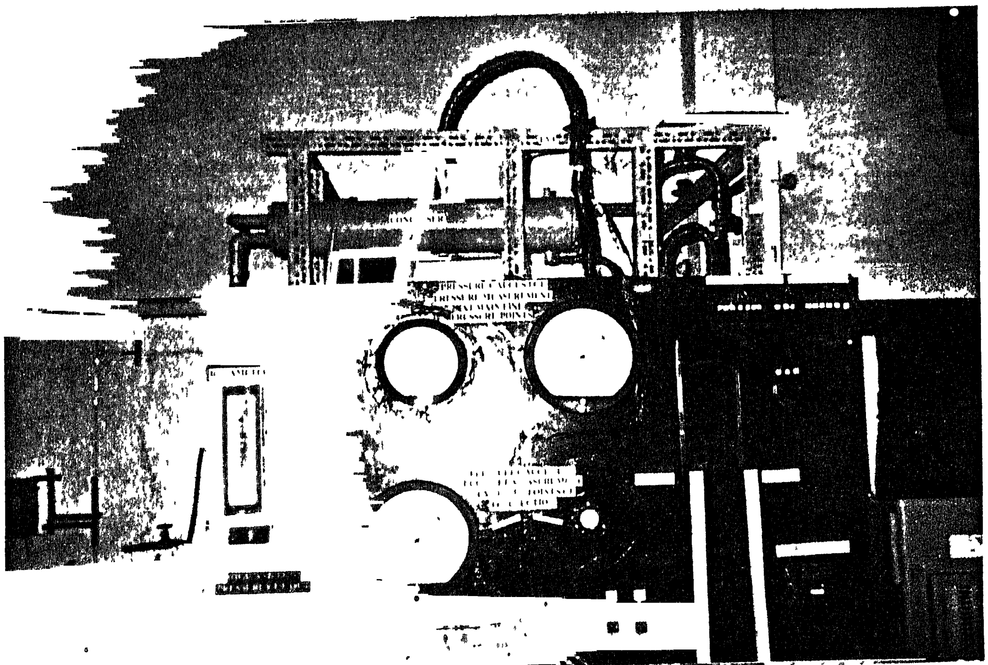


Figure 3.5. Front view photograph of the system with insulation

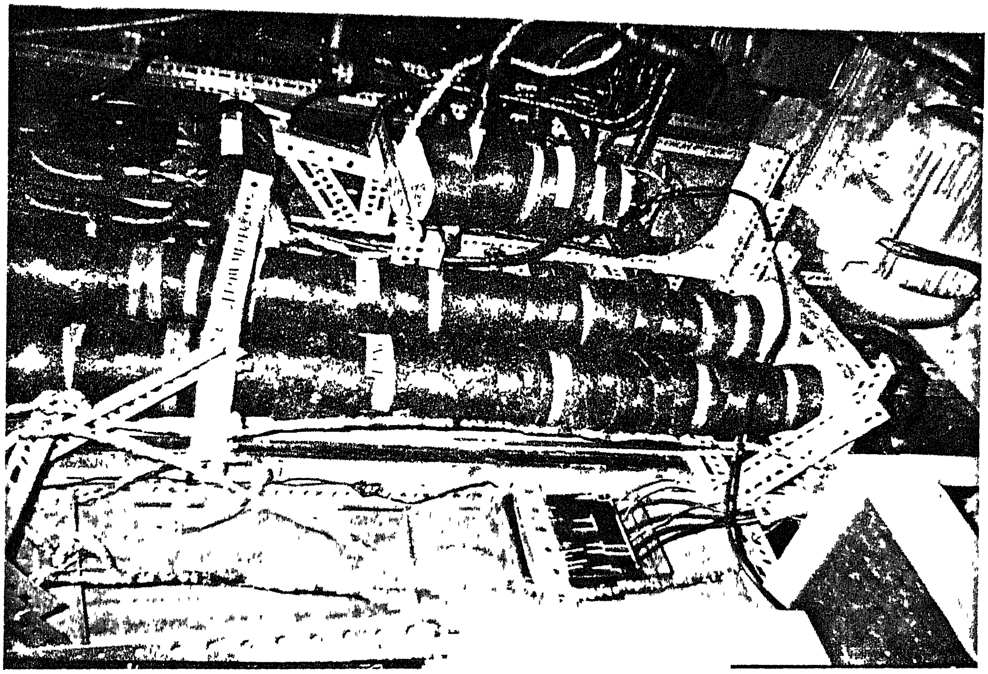


Figure 3.6. Rear view photograph of the system with insulation

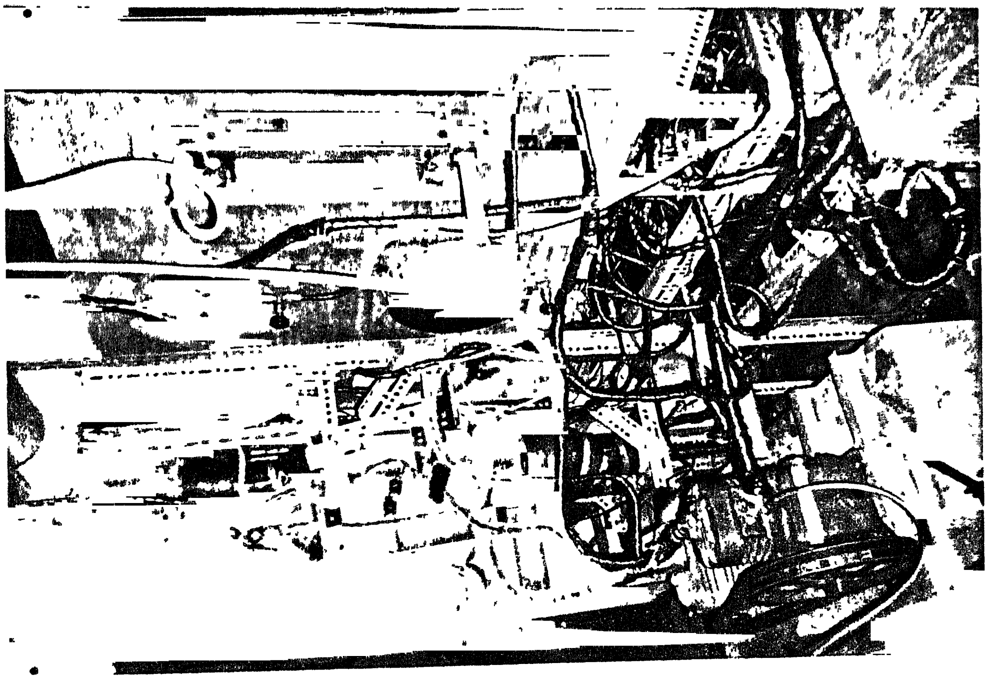


Figure 3.7. Rear view photograph of the system with insulation

test Route 2 is taken. The choice of the route depends on the user and the refrigerant can be made to pass through either of the routes by operating the relevant valve. The refrigerant at the end of the test section is a mixture of liquid and refrigerant, which is now passed through the evaporator coils. The refrigerant vapour obtained from the evaporator, may be wet, saturated or superheated vapor depending upon the surface area of the evaporator coil. If the vapor is wet, it is desirable to pass it through a liquid separator before being fed to the compressor. However, provision of the liquid separator has not been made in this experimental set-up.

3.3 Components, Description and their Fittings

3.3.1 Major Components. The major components used in the setup are as given in table 3.1.

Table 3.1. Major Components of the System

SL. No.	Component
(i)	Oil separator
(ii)	2-Ton, R-12 Compressor
(iii)	Condenser of 2-Ton capacity
(iv)	Receiver
(v)	Flowmeter
(vi)	Drier
(vii)	Thermostatic expansion valve
(viii)	Test sections of various lengths and diameters (described later in chapter 4)
(ix)	Evaporator

3.3.2 Accessories. The various other accessories used in the system are as follows.

Table 3.2. Accessories used in the System

SL. No	Accessory	Qty
(i)	12.7 mm (½") copper pipe	22.86 m (75 ft)
(ii)	6.35 mm (¼") copper pipe	45.72 m (150 ft)
(iii)	15.87 mm (5/8") copper pipe	4.572 m (15 ft)
(iv)	Hoke's needle valve	24 Nos
(v)	6.35 mm (¼") Brass Nuts	90 Nos
(vi)	15.87 mm (5/8") Brass Nuts	07 Nos
(vii)	9.525 mm (3/8") Brass Nuts	02 Nos
(viii)	12.7 mm (1/2") Brass Nuts	27 Nos
(ix)	6.35 mm (¼") Brass 'T's	02 Nos
(x)	2.44 m x 1.22 m (8'x4') Plywood	01 Nos
(xi)	Pressure gauges:- (aa) Up to 300 PSI (20.69 bars) (ab) Up to 50 PSI (3.45 bars) (ac) Up to 100 PSI (6.89 bars)	 02 Nos 01 Nos 01 Nos
(xii)	Electrical Accessories:- (aa) 15 Amps switches (ab) 5 Amps switches (ac) 15 Amps three pin socket (ad) Electrical wire,sheathed, 3 core (ae) Electrical wire ,sheathed, 2 core	 03 Nos 04 Nos 02 Nos 15.24 m (50 Ft) 15.24 m (50 Ft)
(xiii)	Neon bulbs	04 Nos

SL.No	Accessory	Qty
(xiv)	Solenoid valves(initially closed)	04 Nos
(xv)	Electric-motor,3-phase,3.7KW, 5HP, 7.5Ampheres, 1445RPM of Kirloskar electric make.	01 Nos
(xvi)	High and Low pressure Cut out	01 Nos
(xvii)	Starter,3-phase,50Hz,Siemens Make	01 Nos
(xviii)	6.35 mm (¼") Union Brass	13 Nos
(xix)	6.35 mm (¼") Adaptor Brass	44 Nos
(xx)	'T' 6.35 mm (¼")Copper	30 Nos
(xxi)	'T' 12.7 mm (½")Copper	05Nos
(xxii)	6.35 mm (¼") Brass Elbow	02 Nos
(xxiii)	Elbow 15.87 mm (5/8") Copper	01 Nos
(xxiv)	Hand operated valve 12.7 mm (½")	08 Nos
(xxv)	Hand operated valve 15.87 mm (5/8")	01 Nos
(xxvi)	Evaporator Items:- (aa) Single phase 1.5 HP, 900RPM electric motor with condenser. (bb) 1.5 TR AC fans (cc)Cooling coil of 1.5 TR AC	 01 Nos 02 Nos 02 Nos
(xxvii)	'V'Belt, Alaska Company Make ,Type B1973 LP/B76	02 Nos
(xxviii)	MS elbow 12.7 mm (½")	04 Nos
(xxix)	MS socket 12.7 mm (½")	04 Nos
(xxx)	MS pipe 12.7 mm (½")	1.22 m (4 Ft)
(xxxi)	Thick walled capillary 533.4 mm (21") long, 10 mm O.D, 4 mm I.D.	02 Nos

SL.No	Accessory	Qty
(xxxii)	Miscellaneous Items	
	(aa)Energy meter, 3-phase, 4-wire,L&T Make	01 Nos
	(bb) Copper eutectic	01 kg
	(cc) Oxygen Cylinder	01 Nos
	(dd) Nitrogen cylinder	02 Nos
	(ee) LPG Cylinder	01 Nos
	(ff) Screws, nuts, bolts	01 kg
	(gg) Putty	01 kg
(xxxiii)	**Water Connection Items**	
	(aa) Condenser union and socket for water connections	02 Nos
	(bb)Rotameter of 10 GPM	01 Nos
	(cc)PVC Pipe	20 m
	(dd)Pump, 2HP	01 Nos
	(ee)12.7 mm diameter hand operated valve	01 Nos

3.3.3 Description of Components

A detailed description of the various components used in the experimental setup is as given in the succeeding paragraphs.

(a) Compressor. An open type, positive displacement compressor of 2-Ton capacity for R-12 of EVEREST Co. make has been used in the system. Both the inlet and outlet valves of the compressor have got provisions for an additional opening. These additional openings at the inlet and the outlet are used for charging and discharging the system, respectively. The open type compressor has primarily been used so that it can be driven by a separate motor and the motor windings are not affected by the type of refrigerant used. Moreover small compressors of capacity less than 2 TR can be substituted for their performance using different types of refrigerants. Also, a 3-phase electric motor of required capacity and specification was already

available in the heat transfer lab, which could be readily used with an open type of compressor.

(b) Oil Separator. Since most of the refrigerants are readily miscible with oil, an oil separator has been incorporated into the system to separate the oil from the refrigerant vapour coming from the compressor. The oil separator is connected to the compressor by means of 685.81 mm (2'3") of 15.87 mm (5/8") diameter copper tube. One end of the copper tube is fitted to the discharge valve of the compressor and the other end to the inlet of the oil separator. The separated oil in the oil separator could be removed by attaching an extra 6.35 mm (¼") diameter copper tube to the bottom of the oil separator and by opening the valve provided in the separator.

(c) Condenser. A two-pass water cooled condenser of approximately 2-Ton capacity, manufactured by the American refrigeration Company, India, has been used to condense the refrigerant vapour coming from the oil separator. The capacity of the condenser is such that it can deliver subcooled liquid refrigerant at various operating conditions of the plant. Ordinary tap water is used for the condenser cooling. The inlet of the condenser is connected to the outlet of the oil separator by means of 15.87 mm (5/8") diameter copper tube of 2.032 m (6'8") length through a 15.87 mm (5/8") hand operated valve. The condenser outlet is connected to the receiver inlet by means of 711.2 mm (28") length of 9.525 mm (3/8") diameter copper tube.

(d) Flow meter. A flow meter has been provided in the system for measurement of the refrigerant flow rate. The flowmeter consists of two identical limbs of heavy-duty MS pipe, 75 mm I.D and 1.0 m long with flanged ends. Four 12.5 mm thick MS end plates are fixed to the flanges of the flowmeter limbs at both the ends with bolts and nuts arrangement. The flanges are fixed to the limbs by means of a 6.35 mm (¼") thick rubber gaskets of Champion Company make, with araldite applied on both faces to make the joints leakproof. Holes of diameter 12.5 mm are drilled and copper straight tubes of the same diameter are brazed at the centre of each of the end plates to provide inlet and outlet connections to the flowmeter. Further, both the limbs of the flowmeter are connected through a 12.5 mm diameter copper tube of 482.6 mm (1'7") length, at

the top for pressure equalization purposes. The refrigerant lines coming in and out of the flow meter at the top and the bottom respectively are joined to the main line through 'T' junctions referred to as 'T' Junctions 'A' and 'B' respectively for top and bottom of the flowmeter. Solenoid valves named as S1, S2, S3, and S4 are incorporated into each of the lines going into and coming out of the flowmeter limbs between the 'T' junctions and the end plates.

Thick walled capillary glass tubes are incorporated in parallel with each of the flow meter limbs. To fix the capillary gauge glass tube in position, 12.7 mm (½") MS sockets are welded on the two 12.7 mm (½") diameter holes made on the surface of the limb. 50.8 mm (2") long, 12.7 mm (½") diameter MS pipes are screwed on these sockets with 12.7 mm (½") MS elbow on top. The elbow on the top portion of each limb has a 609.6 mm (2') long, 12.7 mm (½") diameter MS pipe screwed in and the bottom side elbow of each limb has 101.6 mm (4") long, 12.7 mm (½") diameter MS pipe screwed in. Between these two open pipes, thick walled capillary of 10 mm O.D and 4 mm I.D (wall thickness-3 mm) and 546.1 mm (21.5") length is inserted. Before inserting the glass tubes all the screwed joints are brazed to rule out the possibility of any leakages. The joints at the glass tube and MS pipes are made leak proof by applying araldite.

The flow meter is connected to the receiver outlet by means of 9.525 mm (3/8") diameter copper tube of 1.6 m (5'3") length and by 12.7 mm (½") diameter copper tube of 533.41 mm (1'9") length through valve DV2 till 'T' Junction 'A' at the top of the flowmeter. There is a bifurcation at the copper 'T' before valve DV2. One end feeds the flowmeter and the other end goes through another valve DV3 meant for discharging purposes.

(e) Drier. A silica gel drier, of UNIWEL make, type UM-163 is fitted in the system to absorb any traces of moisture that might mix with the refrigerant and may cause choking of the test section either partially or completely.

(f) Thermostatic Expansion Valve. A 2-Ton capacity thermostatic expansion valve made by FLICA, Germany, has been used for throttling the refrigerant during the

testing of the setup to find leakages, if any. The sensing bulb of the expansion valve is attached to the wall of the suction pipe at the compressor inlet.

The 12.7 mm (½") diameter copper tube line going out from the drier is bifurcated in two routes by means of 12.7 mm (½") diameter copper 'T' (referred to as 'T' Junction No 1) fitted at a distance of 508 mm (1'8") from the hand operated valve DV4 .The first bifurcation is 330.2 mm (1'1") of 12.7 mm (½") diameter copper tube from 'T' Junction No 1 which leads to a 12.5 mm (½") hand operated valve, DV5 placed for opening or closing of Route No 1. After the valve DV5, 76.2 mm (3") of 12.7 mm (½") copper tube is taken of which one end is connected to the hand operated valve DV5 and to the other end the thermostatic expansion valve is attached. To the discharge end of the T.E.V, a small piece of 15.87 mm (5/8") pipe of 76.2 mm (3") length is attached. To this piece, 12.7 mm (½") diameter copper tube of 736.31 mm (2'5") length is brazed which leads to another hand operated valve DV6 used for controlling flow through this route.

The second bifurcation leads to the capillary test sections, which is Route No 2. There are two hand operated valves DV7 and DV8 placed one after another on this route to enable better control of refrigerant flow through the test sections. After the capillary a valve is again provided to cut off this route whenever desired. The two routes merge in one by providing a 12.7 mm (½") copper 'T' (referred to as 'T' Junction No 2). In this way either of the routes can be opened or closed as per the requirement.

(g) Evaporator. The evaporator has been fabricated in the refrigeration lab itself. Two cooling coils of discarded 1.5 Ton capacity window AC's were taken. The coils were fitted on to a tray of size 762 mm x 762 mm (30"x30") made of 24 gauge Aluminium sheet with approximately 38.1 mm (1.5") of the tray folded from all the four sides to give it a rectangular shape. At the centre of the tray a base of 114.3 mm (4.5") height was welded on which 1.5 HP single-phase motor of 900 RPM with a central shaft of 152.4 mm (6") on either side was fitted. At the two ends of the shaft, a blower/fan of 1.5Ton AC was fitted. These fans were therefore driven by the same motor. Suitable ducts of 24 gauge thick aluminium sheet were made. The cooling coils were then fixed to the ducts by means of 12.7 mm (½") screws and complete assembly then welded to

the tray. The point kept in mind was that since the two fans were driven by the same motor and hence rotated in the same direction. Whereas one blower sucked warm air from the surroundings the second blower drew air which has already passed over the first cooling coil. Besides this, the second coil shall also suck air from within the gap existing between the two coils. Thus the gap between the two coils should be sealed using a 24gauge aluminum sheet and enclose within itself the motor and the shafts. Alternatively, instead of making a duct for each coil separately and then air sealing the space between the two cooling coils, which involves extra effort of welding and cutting of sheets, it was better to make one box which could enclose all the components with one side of the coils left open for sucking the surrounding air and the other for discharging the cold air.

1.219 m (4') of 12.7 mm (½") diameter copper tube line was drawn from 'T' Junction No 2, to the inlet of one of the cooling coils (i.e. to the bottom of the first coil). The two coils were placed in series by brazing 812.81 mm (2' 8") of 9.525 mm (3/8") diameter copper tube connecting the discharge end of the first coil (top end) to the inlet end of the second coil (bottom end). Thus a two phase mixture of liquid and vapour refrigerant enters the cooling coil and wet vapour of the refrigerant leaves the cooling coil. The refrigerant vapour produced in the evaporator enters the suction of the compressor by means of 609.61 mm (24") length of 15.87 mm (5/8") diameter copper tube.

(h) Pipelines. All the piping consists of copper tubing of different diameters and lengths. The length of piping between various points is as given in table 3.3.

3.4 Test Sections

Commercially available three different sizes of copper tubing are used for making capillary tube test specimens. The diameters of these test sections are determined by the procedure as listed in subsequent paragraphs. However, for testing purposes, only one test section has been prepared and connections completed to demonstrate the method to be employed. The size of the capillary used is M90 which is available in the market. To study the entire effects and to generate a reasonable data base for different capillaries

and for different refrigerants, a large number of test specimens will have to be fabricated as discussed in the following chapter.

Table 3.3. Length and Diameter of Copper Pipe Connecting Various Points of the System

SL. No.	From	To	Diameter of Piping	Length of Piping
1	Compressor	Oil separator	15.87mm (5/8")	685.81mm
2	Oil separator	Valve DV1	15.87mm (5/8")	1.321 m
3	Valve DV1	Condenser	15.87mm (5/8")	711.21mm
4	Condenser	Receiver	9.525mm (3/8")	711.21mm
5	Receiver	Copper 'T' Before Valve DV2	9.525mm(3/8") 12.7 mm(½")	1.6 m 254 mm
6	Copper 'T' before Valve DV2	Valve DV2	12.7 mm (½")	76.2 mm
7	Copper 'T' Before Valve DV2	Valve DV3	12.7 mm (½")	1.016 m
8	Valve DV2	'T' Junction 'A' at the top of the flowmeter	12.7 mm (½")	203.2 mm
9	'T' Junction 'A' at the top of the flowmeter	Solenoid valve,S1	12.7 mm (½")	152.4 mm
10	'T' Junction 'A' at the top of the flowmeter	Solenoid valve,S2	12.7 mm (½")	228.6 mm
11	Solenoid valve,S1	Left limb(top)	12.7 mm (½")	228.6 mm
12	Solenoid valve,S2	Right limb(top)	12.7 mm (½")	228.6 mm
13	Length of pressure equalization connection for the flowmeter		12.7 mm (½")	482.61mm
14	Right limb(bottom)	Solenoid valve,S3	12.7 mm (½")	381 mm

SL. No.	From	To	Diameter of Piping	Length of Piping
15	Left limb(bottom)	Solenoid valve ,S4	12.7 mm (½")	381 mm
16	Solenoid valve,S3	'T' Junction 'B' at the bottom of the flow meter.	12.7 mm (½")	127 mm
17	Solenoid valve,S4	'T' Junction 'B' at the bottom of the flow meter.	12.7 mm (½")	127 mm
18	'T' Junction 'B' at the bottom of the flow meter.	Valve DV4	12.7 mm (½")	228.6 mm
19	Valve DV4	Drier	12.7 mm (½")	76.2 mm
20	Drier	'T' Junction No 1	12.7 mm (½")	431.81mm
21	'T' Junction No 1	Valve DV5	12.7 mm (½")	330.2 mm
22	Valve DV5	Thermostatic expansion valve	12.7 mm (½")	76.2 mm
23	Thermostatic expansion valve	Valve DV6	15.87 mm(5/8") 12.7 mm (½")	76.2 mm 736.61mm
24	Valve DV6	'T' Junction No 2	12.7 mm (½")	50.8 mm
25	'T' Junction No 1	Valve DV7	12.7 mm (½")	508 mm
26	Valve DV7	Valve DV8	12.7 mm (½")	279.4 mm
27	Valve DV8	Valve DV9 (Through the test section)	Capillary Test Sec.& small lengths of 12.7 mm (½")and 6.3 mm (¼") copper tube	
28	Valve DV9	'T' Junction No 2	12.7 mm (½")	1.092 m
29	'T' Junction No 2	Evaporator inlet	12.7 mm (½")	1.219 m
30	Evaporator outlet	Compressor inlet	12.7 mm (½") 15.87 mm(5/8")	152.4 mm 609.6 mm

3.4.1 Connections to the test section. The test sections are to be connected between valves DV8 and DV9. The test section is laid in the desired shape between the two valves and then the two free ends are connected to the respective valves by small lengths of 6.35 mm, 9.525 mm (¼", 3/8") and then finally by a 12.7 mm (½") diameter copper pipes. All the pipes are brazed together to give leak proof joints. The free end of 12.7 mm (½") copper pipe is flared and fitted with a 12.7 mm (½") brass nut which is finally screwed on to the respective valves at the two ends.

3.4.2 Finding the Internal Diameter of the Capillary Test Section. The internal diameter of the capillaries can be found out by the following method. An accurately measured length of the capillary is taken and one of its ends is sealed. Distilled water is poured in the capillary till it fills it up completely. The water is then emptied in a beaker and the quantity is accurately measured in milli litres and converted in units of cubic metre. This would be the volume of the capillary of a specific length. When equated to

$\pi/4 \times D \times D \times L$ = Volume of the capillary, where,

D=Diameter of the capillary (to be found out),

L =Length of the capillary (already known),

would give the internal diameter. However, a more reliable method is to measure the diameter with an accurate vernier calliper as has been done for few of the commercially available capillary tubes. The diameters of three such tubes are as follows:-

(a) M90 - O.D = 3.18 mm, I.D = 2.18 mm

(b) M85 – O.D = 2.88 mm, I.D = 1.88 mm

(c) M70 – O.D = 2.05 mm, I.D = 1.05 mm

The internal and outer diameter of the capillary tubes is required to firstly, specify the exact size and secondly, to find the diameter of the wooden mandrill for making helical and spiral test sections as discussed later.

The specifications of the vernier calliper used are as follows:-

(a) Type – Dial gauge vernier caliper,

(b) Make – MITUTOYO (Japan),

(c) Maximum Range – 200 mm,

(d) Least Count - .05 mm.

3.5 Instrumentation and Control

3.5.1 Measurement of Refrigerant Flow Rate

The refrigerant flow is measured by means of a flowmeter. Both the limbs of the flowmeter are used simultaneously during the experimentation; while one supplies a measured quantity of the refrigerant through driei to the expansion device, the other limb keeps on collecting the condensate at the same time. The flowmeter is required to be calibrated as per the procedure given below.

(a) **Calibration of the limbs.** Accurately measured volume of double distilled water is poured into one of the flowmeter limbs and the gauge glass is graduated accordingly. The process is continued till it completely fills the flowmeter limb. Same procedure is repeated for the graduation of the second flowmeter limb. The graduations can be marked in litres. Each limb of the flowmeter is found to have a capacity of 4.75 litres. Alternatively, there is no requirement of calibration of the limbs since the internal diameter of the limbs is known. The use of the limbs is to determine the flow rate of the refrigerant for which the fall in height of the refrigerant in either of the limbs is required to be known. This can be done by means of placing 2 Nos, 60 cms long plastic scales in inverted position next to the glass pipes of the flow meter limb. This can be used as the calibration scale for measurement of the flow rate of the refrigerant as explained in subsequent paragraph.

(b) Charging of the Flow meter. As said earlier, both the limbs are used during the experimentation. The system is initially charged by keeping solenoid valve 'S1' open and valves 'S2','S3', and 'S4' closed till such time the first limb is completely filled. As soon as the first limb gets filled solenoid valve 'S1' is closed and valves 'S2', 'S4' is opened. Valve 'S3' is allowed to remain closed. This would allow charging of the system ahead of the flowmeter via limb'A' whereas the limb 'B' would receive refrigerant from the condenser. By the time limb 'A' gets emptied, limb'B' would have got filled. By opening valves 'S1' and 'S3' and by closing valves 'S2' and 'S4' limb 'A' will get filled and limb 'B' shall charge the system ahead of the flow meter. In this

manner the system can be continuously operated by alternately opening and closing (simultaneously) the two sets of diagonal valves namely 'S1 '& 'S3' and 'S2' &'S4'.

(c) **Mass or volume flow rate.** Mass or volume flow rate of the refrigerant can be found out during the experiment itself. Say, one of the limb has been completely filled and has started discharging downstream of the flow meter. At any point of time t1, the level of the refrigerant in a particular limb is marked as x1. At any other instant t2, the level of the refrigerant would drop to, say x2. Levels x1 and x2 can be directly read from the calibration scale. The volume flow rate of the refrigerant would then be given as follows

$$\text{Volume flow rate (litres/sec)} = [(\pi/4) \times D \times D \times (x1-x2)] / (t2-t1)$$

To find the mass flow rate, the above value can be multiplied by the density of the refrigerant at condenser temperature.

3.5.2 Pressure measurements

The pressure measurement is done through four pressure gauges of different ranges appropriate to high and low pressures in the system and the test section.

(a) Calibration of the pressure gauges. The pressure gauges are calibrated using an Amthor dead weight tester and weights, standardized by the National Bureau of standards, U.S.A. The gauges are found to be accurately calibrated as no difference in the readings of the gauge and the applied loads could be discerned.

(b)Range of Pressure Gauges used. All in all four pressure gauges of ranges as shown below were used in the setup to determine pressures at six locations in the main line and at 13 points on each of the test sections. The pressure gauges are as follows

- 300 PSI (20.69 bars) – 2 Nos.
- 100 PSI (6.89 bars) – 1 Nos.
- 50 PSI (3.45 bars) – 1 Nos.

One 300 PSI and one 50 PSI pressure gauge was used for determining pressure at the six locations in the main line. The circuitry connecting the points to the pressure gauges is as shown in figure No 3.8. The points on the main line are as follows

- Inlet of the compressor
- Exit of the compressor
- Inlet to the condenser
- Exit of the condenser
- Exit of the flow meter/inlet of the expansion device (or the test section)
- Exit of the test section or inlet of the evaporator

One 300 PSI and one 20 PSI pressure gauge were used for measuring pressures at 13 points in the test section. The circuitry connecting the points to the pressure gauges is as shown in figure No 3.9.

(c) Connections of pressure tapping

(i) Valves. A Hoke's needle valve is fitted in the line of each pressure tapping as shown in the figures 3.8 and 3.9. Valves V1 toV13 are used for the test section tappings and valves V16 to V22 are used for the main line pressure tapping. Valves V14, V15, V23 and V24 are connected to the pressure gauges to cut them off when required. The valves V16 and V17 are used on low-pressure side and rest of the valves i.e.; V18 to V22 are for high-pressure side. Thus the relevant pressure gauge valve in accordance with the Hoke's valve should be opened. Valve V14 and V23 are the low pressure gauge valves whereas V15 and V24 are high pressure gauge valves. The various valves represent different points of the system as listed in table 3.4.

For example, if we wish to find pressure at the inlet of the compressor then all other valves on this circuitry are kept closed except valves V17 (connecting the point) and valve V23 (valve with low pressure gauge). Valves V1 to V13 are connected to different points on the test section. Points are at specified distances of 30 cms each starting from the drier side and proceeding towards the evaporator side and have been numbered accordingly i.e. valve V1 represents point on the test section closest to the drier and valve V13 represents point closest to the evaporator.

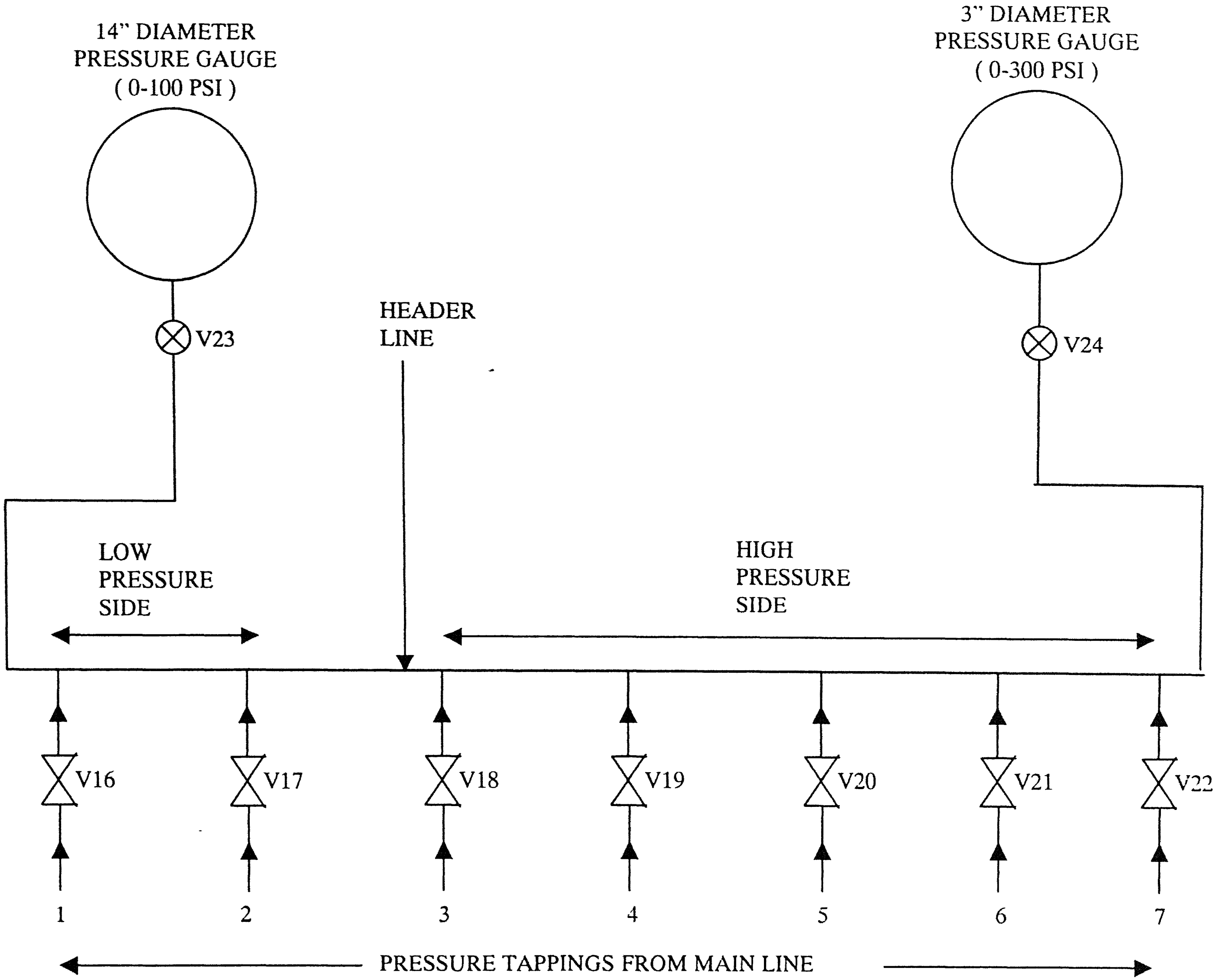


Figure 3.8 Line Diagram for Pressure Measurements at Different Points on the Main Line

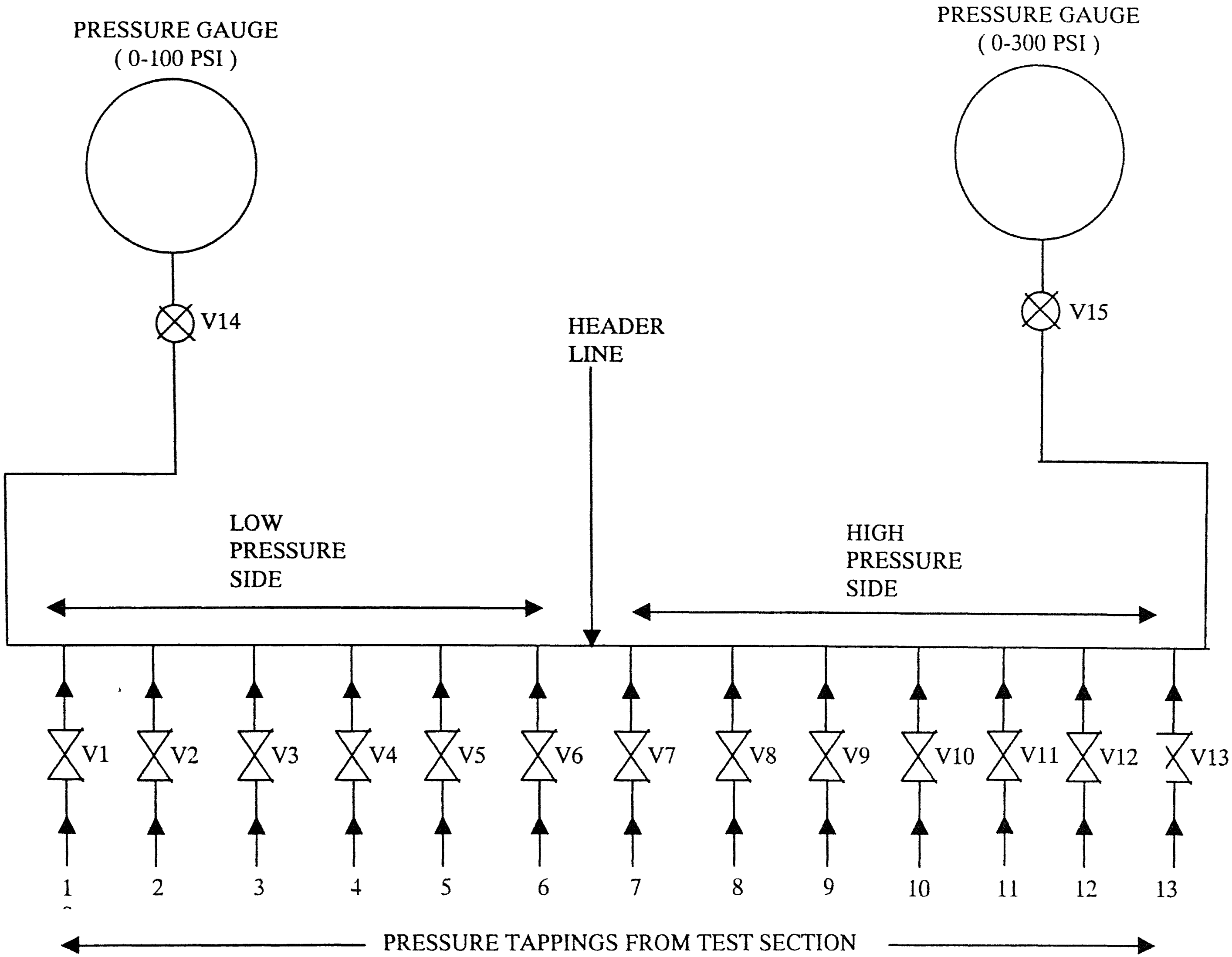


Figure 3.9 Line Diagram For Pressure Measurements At Different Points On The Test Sections

Table 3.4. Nomenclature of Hoke's Needle Valves Used in the System for Pressure Measurements

SL. No	Corresponding Place	Nomenclature Of The Hoke's Valve
	At the test section	
(i)	Points 1 to 13 starting from the drier side to the evaporator side.	V1 to V13
(ii)	Low pressure gauge of test section pressure measurement circuitry.	V14
(iii)	High pressure gauge of test section pressure measurement circuitry	V15
	At the Main Line	
(i)	Exit of the test section or inlet of the evaporator	V16
(ii)	Inlet of the compressor	V17
(iii)	Exit of the compressor	V18
(iv)	Inlet to the condenser	V19
(v)	Exit of the condenser	V20
(vi)	Exit of the flow meter/Inlet of the expansion device(or the test section)	V21
(vii)	SPARE	V22
(ix)	Low pressure gauge of main line pressure measurement circuitry	V23
(x)	High pressure gauge of main line pressure measurement circuitry	V24

(ii) **Main line pressure tapping.** In the mainline, the place where it is desired to find the pressure, a hole of 4 mm diameter is drilled. A small length, approximately 76.2 mm (3") of 6.35 mm (¼") diameter copper tube is taken and a 6.35 mm (¼") copper 'T' is brazed at one of its ends. This assembly is then brazed onto the hole drilled in the main line. The end to which the copper 'T' is brazed shall have two free

openings. To one of the openings, the line for pressure tapping is brazed. The second open end is closed by means of brazing a small length of 6.35 mm (¼") copper tube whose one end is choked except for points before and after the compressor. At these two points the connections to the high and low pressure cut out is made as explained in section 3.5.9.

A suitable length of 6.35 mm (1/4") diameter copper tube, as measured on ground and as listed in Table 3.5, is taken with one of its end flared. The unflared end of this copper tube is brazed to one of the openings of the copper 'T' at the pressure hole. The flared end of this line is connected to the respective Hoke's valve which controls pressure measurement at that point. Another line of 6.35 mm (¼") copper tube is taken from that Hoke's valve to the header line connecting the pressure gauges as shown in figures 3.8 and 3.9. The distances and locations of these pressure tappings on the main line are given in Table No 3.5. This also gives out the length of the copper tube used for connecting the point to the respective Hoke's valve though it is absolutely immaterial and can be increased or decreased by increasing or decreasing the number of turns.

Alternatively, instead of using 6.35 mm (¼") copper 'T', a 6.35 mm (¼") brass elbow can be used for connecting the pressure holes to the respective Hoke's valve. A small length of 101.6 mm (4"), 6.35 mm (1/4") diameter copper tube with one end flared and fitted with 6.35 mm (¼") brass nut is brazed at the pressure holes. To the flared end of this small length, a 6.35 mm (¼") brass elbow is fitted. Suitable length of 6.35 mm (¼") copper tube with both ends flared and fitted with 6.35 mm (¼") brass nuts is taken and attached to the 6.35 mm (¼") brass elbow at the pressure hole and the respective Hoke's valve at its other end.

(iii) Test Section Pressure Tappings. In a similar manner as above, pressure tappings for the test section are provided. However, since the diameter of the test section is very small, it is not possible to drill holes or carry out any brazing work after fixing it on the set-up. Moreover, for hastening up the work, drilling of holes and brazing work was done in the test section before assembling it in the setup. The procedure is as described below. A test section of desired diameter and length is taken.

Table 3.5. Distances of Pressure Tappings from various Components and Length of Copper tube From Pressure Tapping points To the Hoke's Needle valve

SL. No.	Pressure Tapping	Distance From	Distance	Length Of Cu Tube (From The Pressure Tapping To Respective Hoke's Valve)
1	Compressor inlet,P2	.Before Compressor inlet .After Evaporator outlet	228.6 mm 956 mm	914.4 mm
2	Compressor exit,P3	.After compressor exit . Before oil separator inlet	127 mm 558.81 mm	2.311 m
3	Condenser inlet,P4	. Before Condenser inlet . After valve DV1 outlet	101.6 mm 609.61 mm	2.184 m
4	Condenser outlet,P5	. After Condenser outlet . Before receiver inlet	635 mm 76.2 mm	1.6 m
5	Flow meter outlet,P6	. After drier . Before 'T' Junction No 1	76.2 mm 355.6 mm	1.575 m
6	Evaporator inlet,P1	. After 'T ' Junction No 2 . Before evaporator inlet	76.2 mm 1.143 m	609.61 m

Points where the pressure is required to be measured are marked. Very minute holes of approximately ¾th of the diameter of the test section are drilled at all these places. To make sure that the holes drilled are free from burrs, a steel wire of a diameter slightly less than the bore of the capillary tube is slowly pushed through the tube a number of times. In addition, a small needle is also used to clean the holes individually. This method of cleaning burrs, which was found to be quite effective, was perfected by carrying out tests on a number of small pieces of capillary tubing, then breaking them open at the holes and carrying out a visual inspection.

76.2 mm to 101.2 mm (3" to 4") length of 6.35 mm (¼") diameter copper tube is taken and a 3 mm long groove of the size of the diameter of capillary is made at its bottom. This tube is placed on top of the hole drilled in the capillary and

tightened with the help of crow spanner. Care is taken so that the hole of 6.35 mm (1/4") diameter tube is directly on top of the hole of the capillary .The 6.35 mm (¼") copper tube is then brazed at that point with the help of copper eutectic. To ensure that the pressure holes do not get choked by molten copper during the above process, high pressure air was made to flow through the capillary test sections. The free end of these 101.2 mm (4") long, 6.35 mm (¼") diameter copper tubes are further brazed to 101.2 mm (4") length of 9.525 mm (3/8") diameter copper tubes. In this manner all the desired test sections are prepared before fitting in the setup. They are now required to be fitted to the respective Hoke's valve, which are further connected to the pressure gauges.

From each of the Hoke's valve (i.e Hoke's valve number 1 to 13), approximately 304.8 mm (12") length of 6.35 mm (¼") diameter copper tube is drawn whose both ends are flared and fitted with 6.35 mm (¼") brass nuts. One end of this line is fitted to the Hoke's valve and the second end is fitted to 6.35 mm (¼") union. The free end of the pressure tappings at the test section are connected to the respective union by means of suitable length of 6.35 mm (¼") diameter copper tube. One end of this tube is flared and fitted with 6.35 mm (¼") brass nut (then fitted to the union) and the second end is inserted and brazed in 9.525 mm (3/8") diameter tube at the free end of the pressure tapping. The test sections are connected to the main line, at their free ends, through small lengths of 6.35 mm (1/4") diameter tubing, followed by 12.7 mm (1/2") diameter tubing with the help of flare fittings.

3.5.3 Temperature Measurements. The temperature is also measured at the same locations where pressure is measured. In addition to this, the atmospheric temperature and the temperature of the refrigerant at the inlet to the test section are also recorded using 24 gauge Copper – constantan thermocouples.

(a) Calibration of thermocouples. The thermocouples are calibrated in the range of -10 to 80 °C before they are fitted in the system. The system employed for the calibration of the thermocouples can be separate for the two temperature zones i.e., for -10 °C to ambient temperature and for ambient temperature to +80 °C.

For calibration of the thermocouples in the temperature zone of ambient temperature to +80 °C, the system used and the method employed is as under. A bath filled with distilled water is taken with arrangement of a heater, a stirrer and a data logger with an accuracy of 0.1°C. The RTD probe of the data logger is used to indicate the actual temperature of water in the bath. The RTD probe is immersed in water to the required depth. The bead of the thermocouple is also suspended up to the same depth as the tip of the RTD probe and kept in its very near vicinity. The two ends of the thermocouple bead are then attached to one of the terminals of a Honeywell 48-point temperature recorder. The temperature of the distilled water initially would be the ambient temperature prevailing at that point of time and shall be shown by the data logger. On pressing the relevant terminal switch to which the thermocouple has been attached the 48-point recorder shall indicate the temperature shown by the thermocouple. For measuring the temperature shown by the thermocouple above the ambient, the system is switched on after fixing the desired temperature setting on the arrangement provided in the calibration bath. Stirrer ensures a uniform temperature of the distilled water in the bath. The flash light provided with the system switches off and indicates that the desired temperature of water has reached. Now the readings from the data logger indicating actual temperature of the distilled water and reading from the 48-point recorder indicating the temperature shown by the thermocouple are noted. In a similar fashion all readings above the ambient temperatures are taken.

For calibrating the thermocouples below the ambient temperature, a separate bath of nearly 2 litres capacity is taken and the RTD probe of the data logger and the thermocouple are suspended as was done for the above mentioned system. Ice is added in steps to lower the temperature of the distilled water and the actual temperature of the distilled water is monitored. As and when the desired temperature is reached both the readings are taken. This process can be continued till a lower limit of approximately 5°C after which to achieve lower temperatures certain quantity of salt is added to the distilled water. The temperature of the distilled water is lowered in steps of 5°C at a time and corresponding readings obtained. A refrigerated bath with a heater control could also be used.

Thus a data base is created for actual temperature versus the temperature shown by the thermocouples. Graphs are drawn for each thermocouple. These are the

calibration curves for the thermocouples which shall be required at the time of processing the temperature data after the experiments. In the experimental setup, the thermocouples are connected to different terminals of the 48-point recorder through the connecting leads. On pressing the respective terminal switch, the instrument indicates the temperature as shown by the thermocouple. However that shall not be the actual temperature of the fluid. This reading will have to be compared with the calibration curve which shall give the actual temperature at a particular point at that instant.

Calibration curves for all the thermocouples are given in figures 3.10 to 3.29.

(b) Measurement of temperature

(i) In the main line. For measurement of temperature in the main line at six different points, following procedure has been adopted. A 152.4 mm (6") long, 6.35 mm (¼") diameter copper tube is taken and bent in 'L' shape with one of its ends choked. A suitable diameter hole is drilled in the main line in which this 'L' shape tube is inserted (with its closed end inside) and then brazed to give a thermocouple well. This thermocouple well is filled with silicon oil and the thermocouple is dipped inside as much as possible to give the accurate measurement of temperature of the refrigerant flowing through that point. This method is suitable for places where the main line is of 15.87 mm (5/8") or minimum of 6.35 mm (½") diameter copper pipe. However, for places where the main line is of 9.525 mm (3/8") diameter copper tube this method may not be suitable since the 6.35 mm (¼") diameter copper line if inserted in 9.525 mm (3/8") copper tube may result in blockage of flow. Thus a different procedure is followed for temperature measurement at the exit of the condenser. For temperature measurement at this point, a 101.2 mm (4") length of 9.525 mm (3/8") diameter copper tube is taken with one of its ends choked. A 76.2 mm (3") long, 15.87 mm (5/8") diameter copper tube, with one of its ends choked is taken and the 9.525 mm (3/8") diameter tube with its closed end is inserted in it. Now the 15.87 mm (5/8") diameter tube's free end is also closed by brazing and a 9.525 mm (3/8") diameter hole is drilled almost in the center of its length. A 9.525 mm (3/8") copper 'T' is brazed at this hole in vertical position. The line connecting the outlet of the condenser is brazed to the top free end of this copper 'T' and the line going to the receiver is brazed to the bottom free

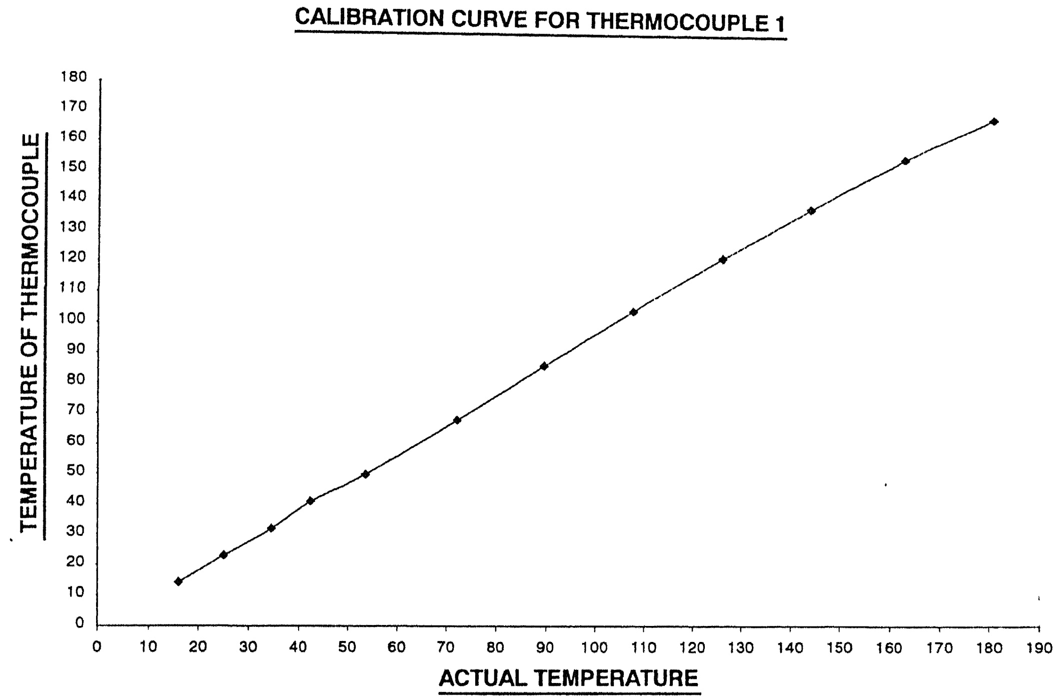


Figure 3.10 Calibration Curve for thermocouple'T1'

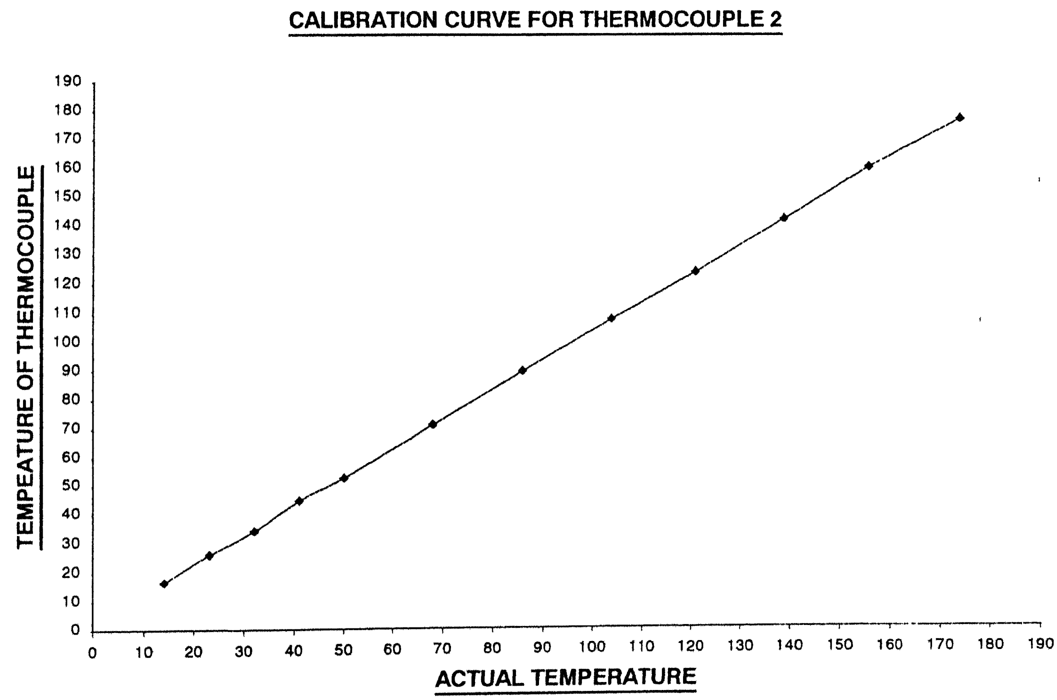


Figure 3.11 Calibration Curve for thermocouple'T2'

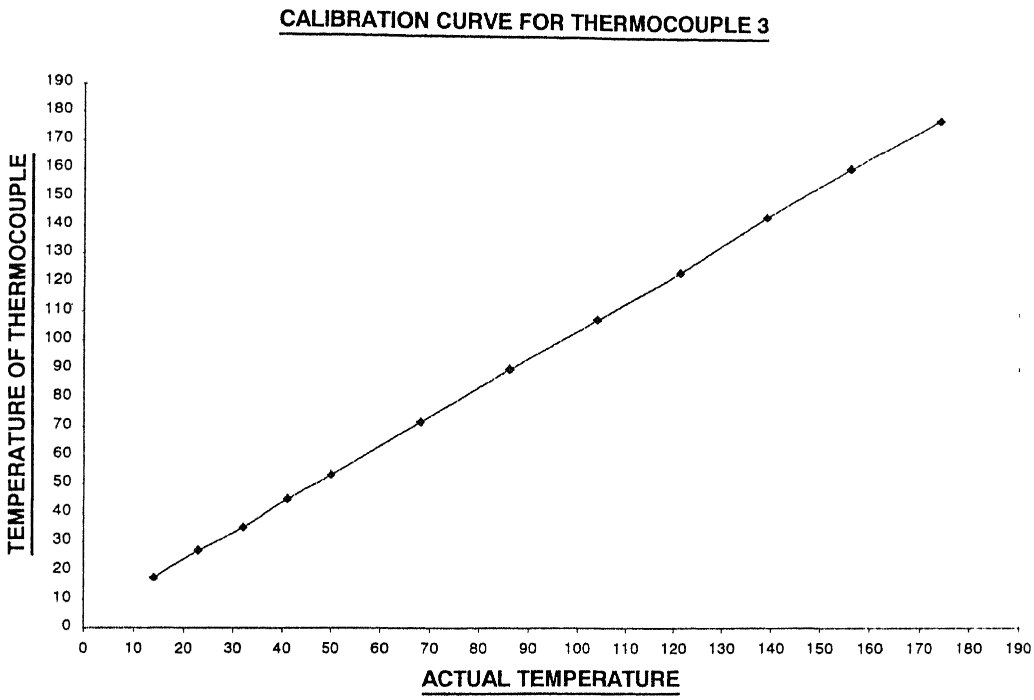


Figure 3.12 Calibration Curve for thermocouple'T3'

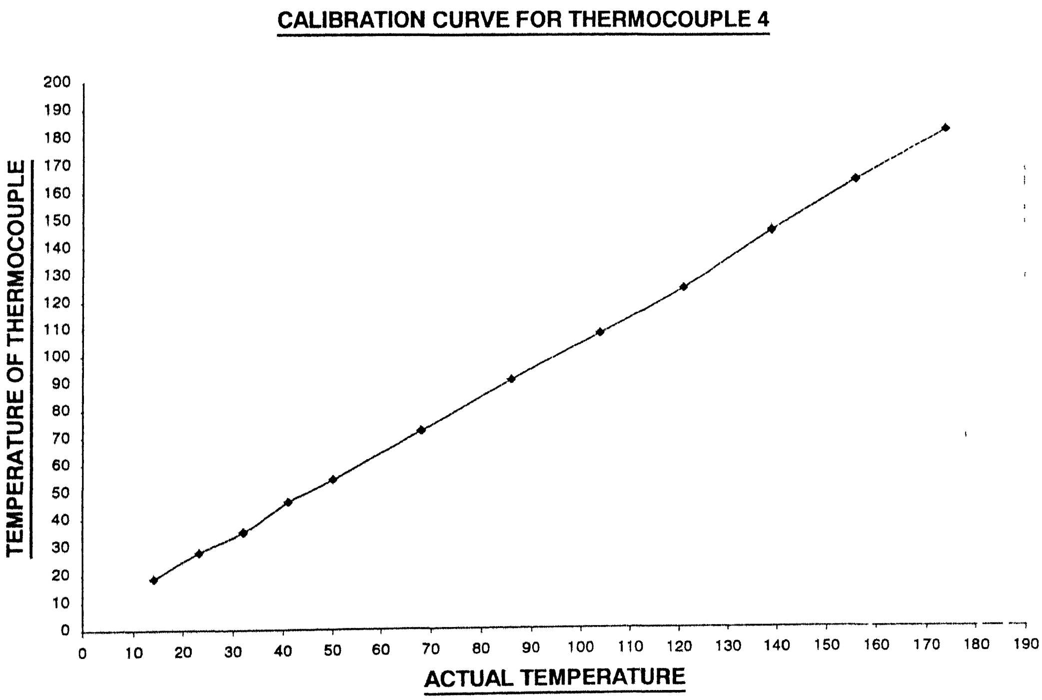


Figure 3.13 Calibration Curve for thermocouple'T4'

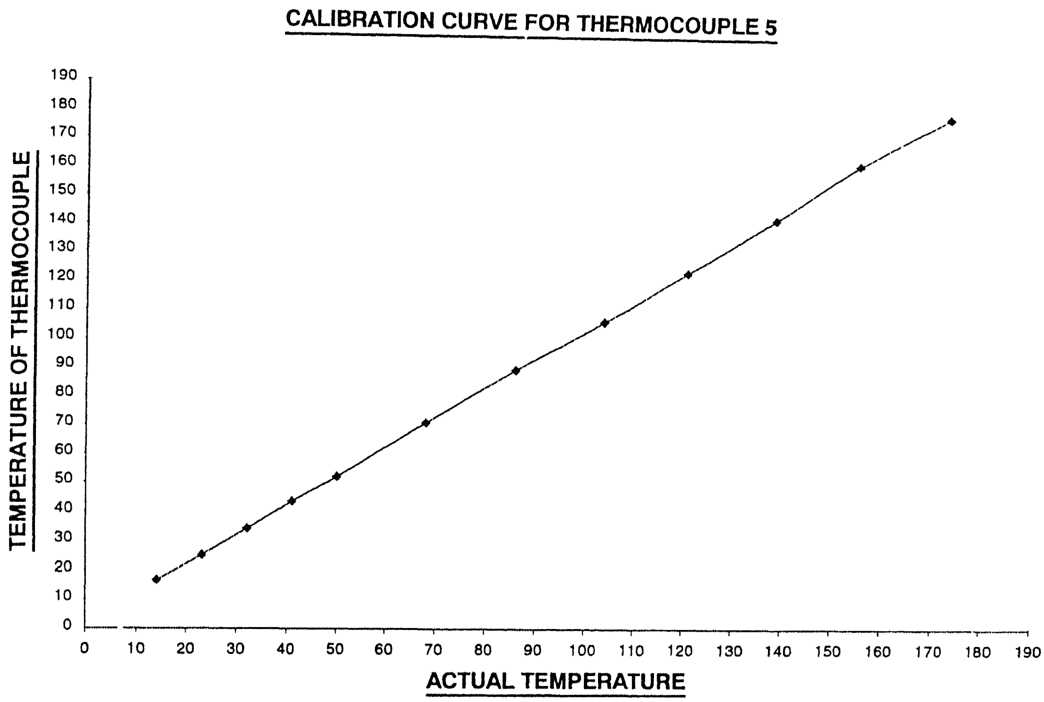


Figure 3.14 Calibration Curve for thermocouple'T5'

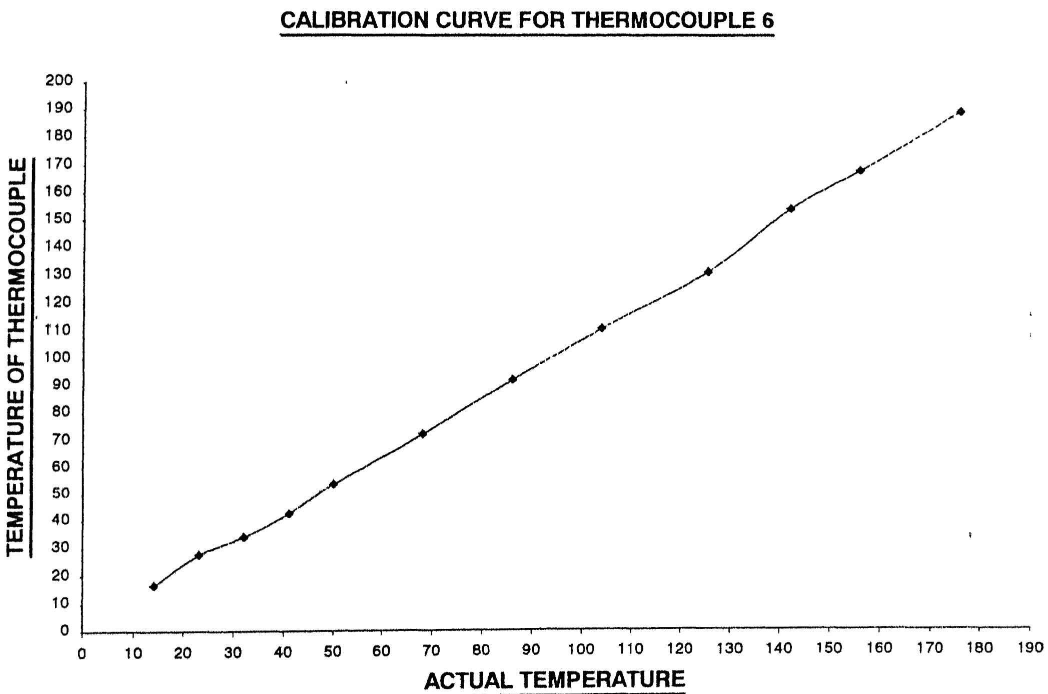


Figure 3.15 Calibration Curve for thermocouple'T6'

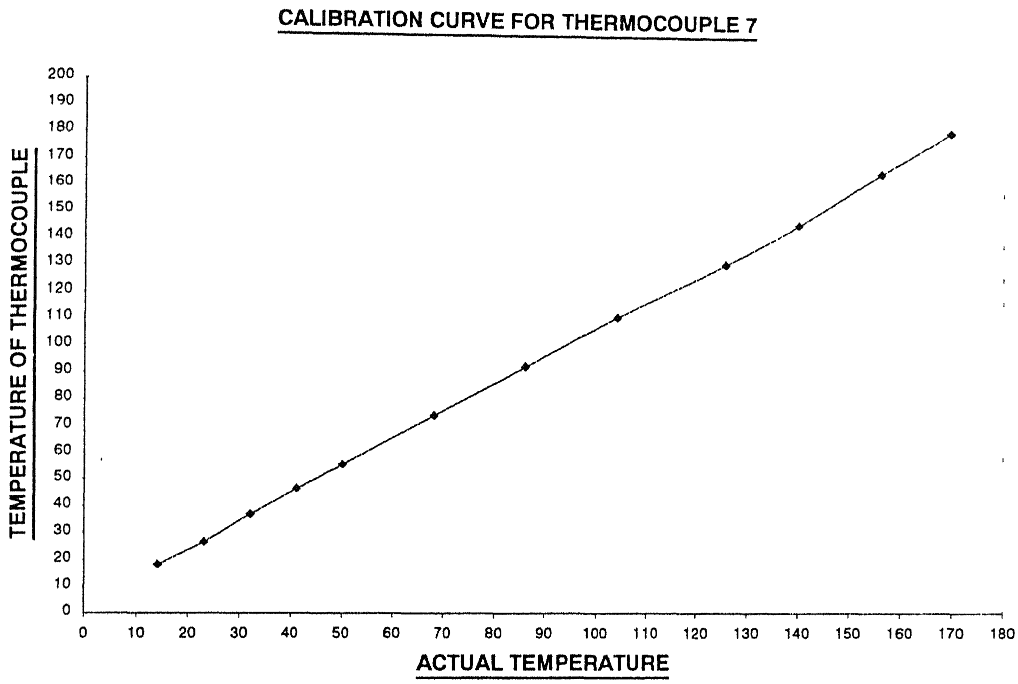


Figure 3.16 Calibration Curve for thermocouple'T7'

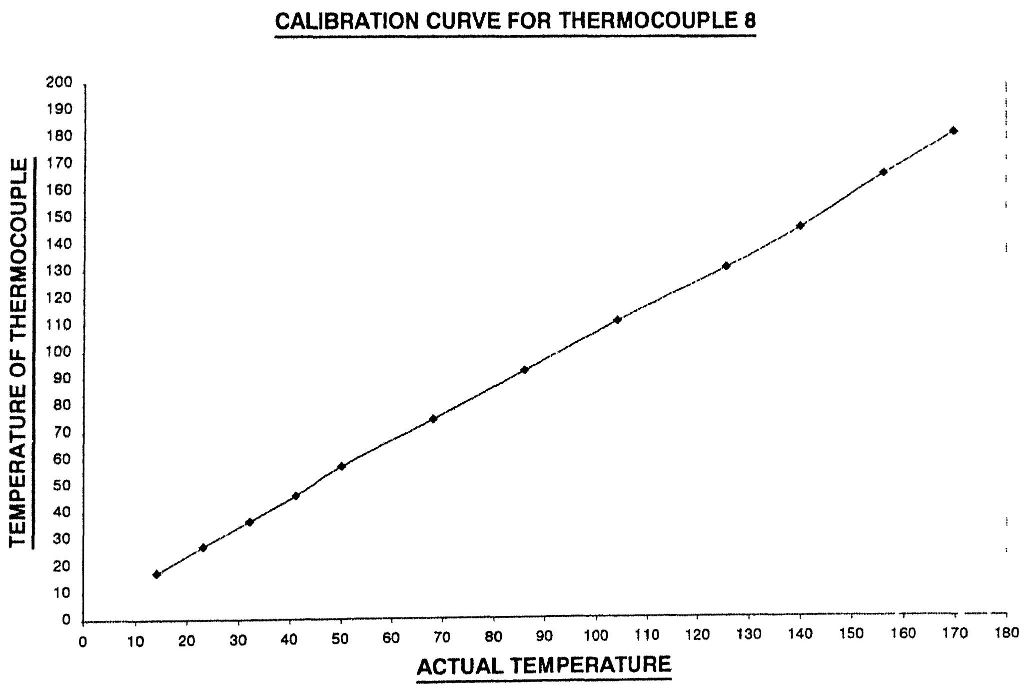


Figure 3.17 Calibration Curve for thermocouple'T8'

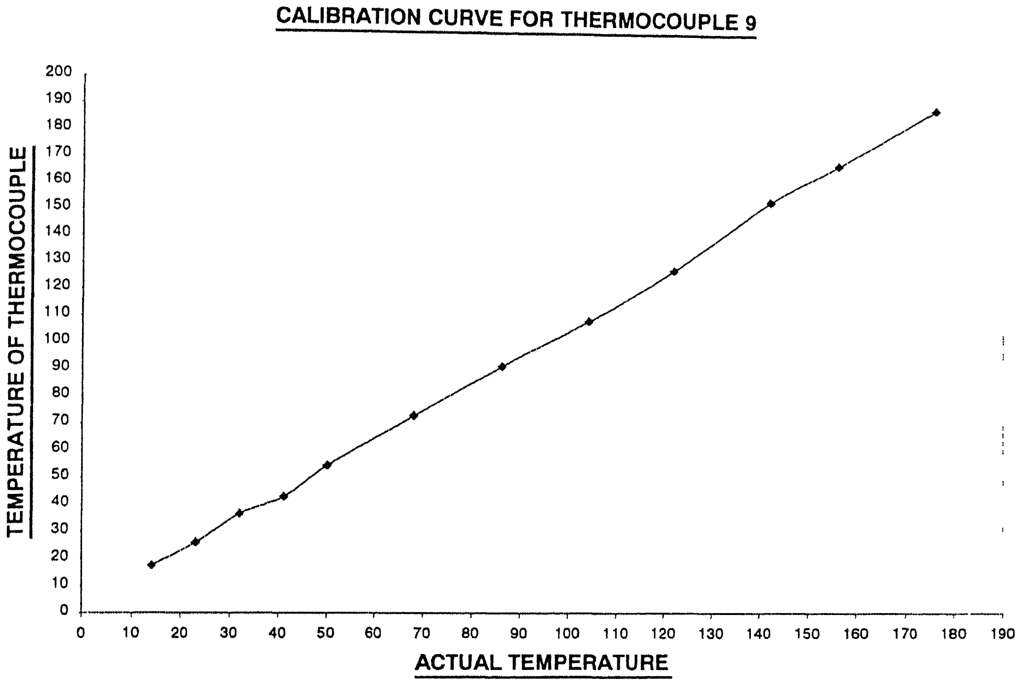


Figure 3.18 Calibration Curve for thermocouple'T9'

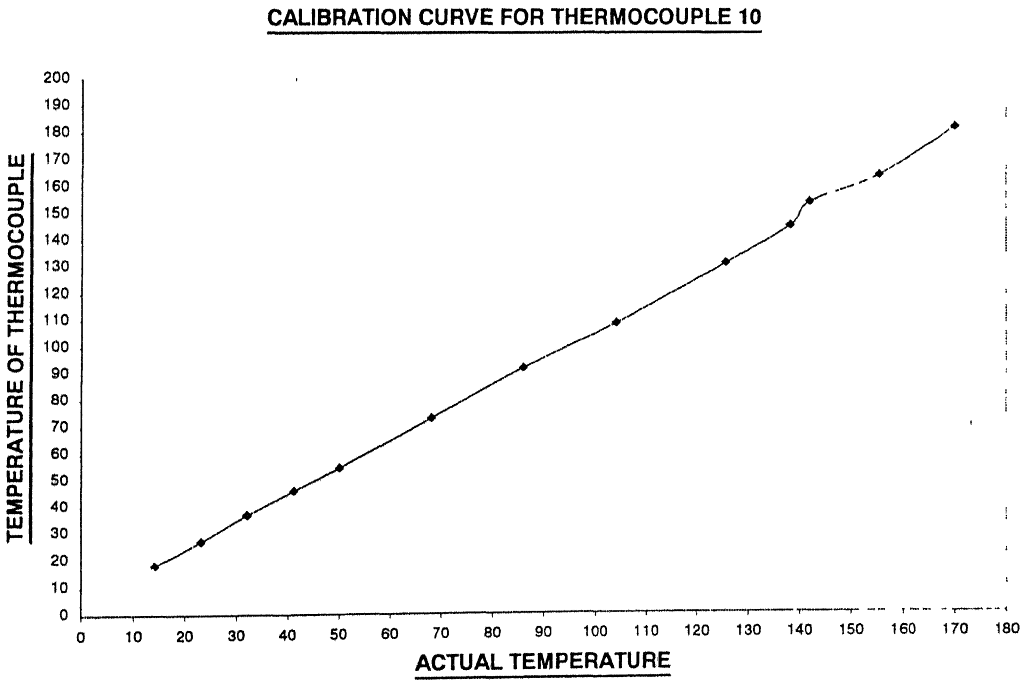


Figure 3.19 Calibration Curve for thermocouple'T10'

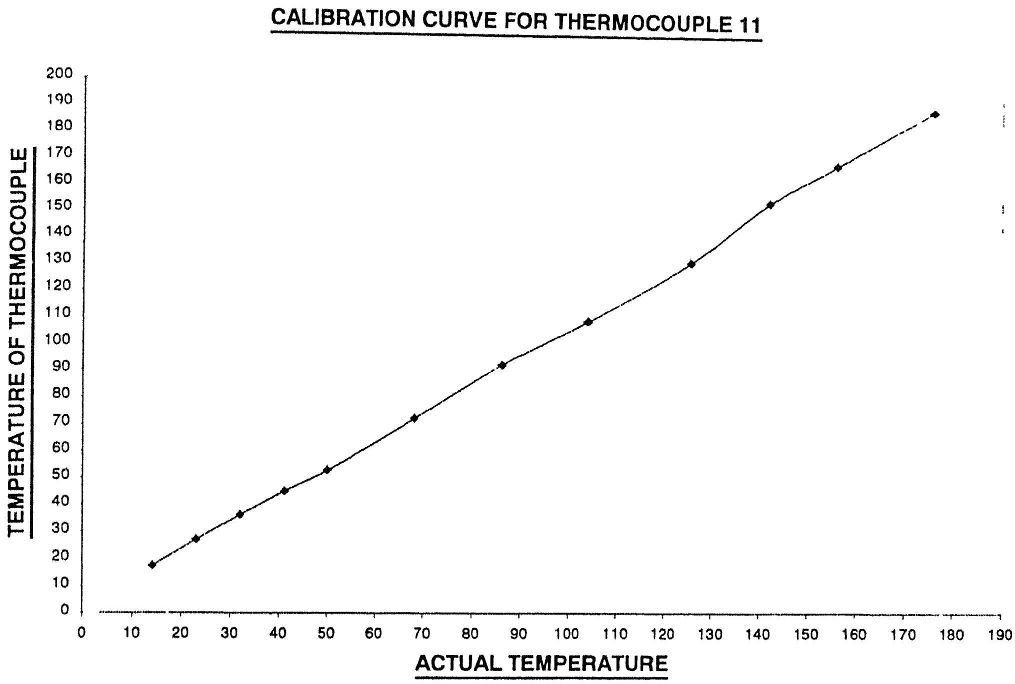


Figure 3.20 Calibration Curve for thermocouple'T11'

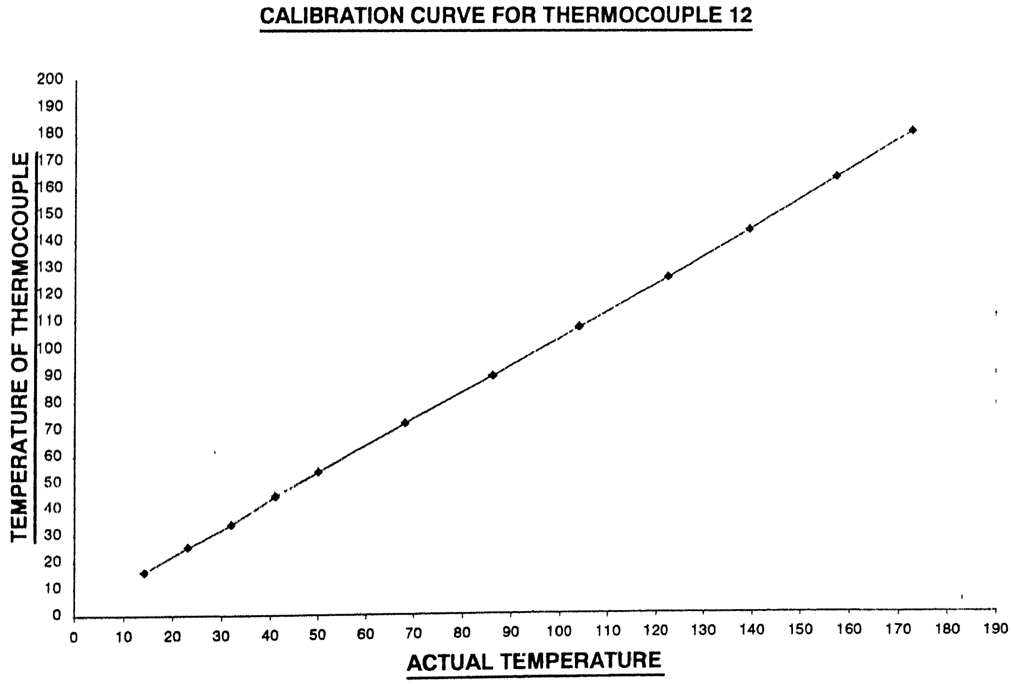


Figure 3.21 Calibration Curve for thermocouple'T12'

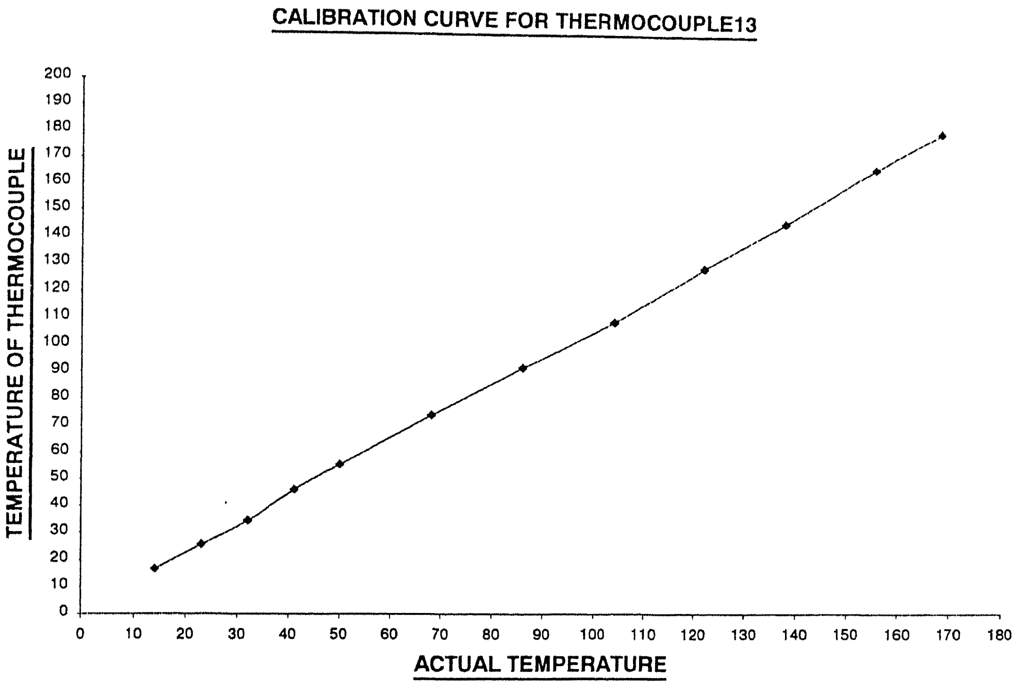


Figure 3.22 Calibration Curve for thermocouple'T13'

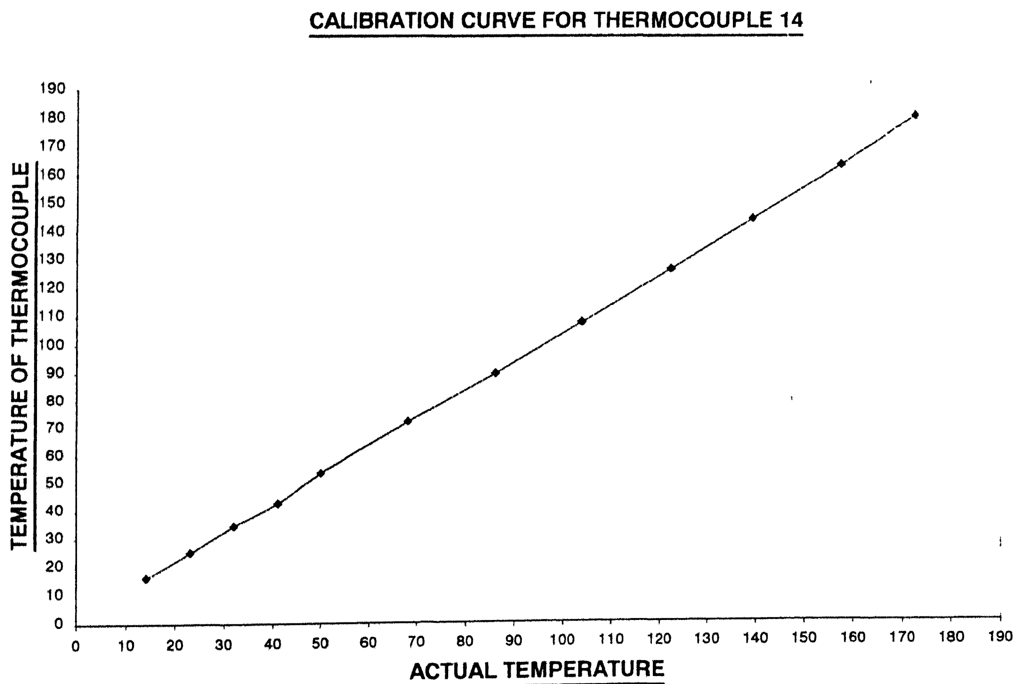


Figure 3.23 Calibration Curve for thermocouple'T14'

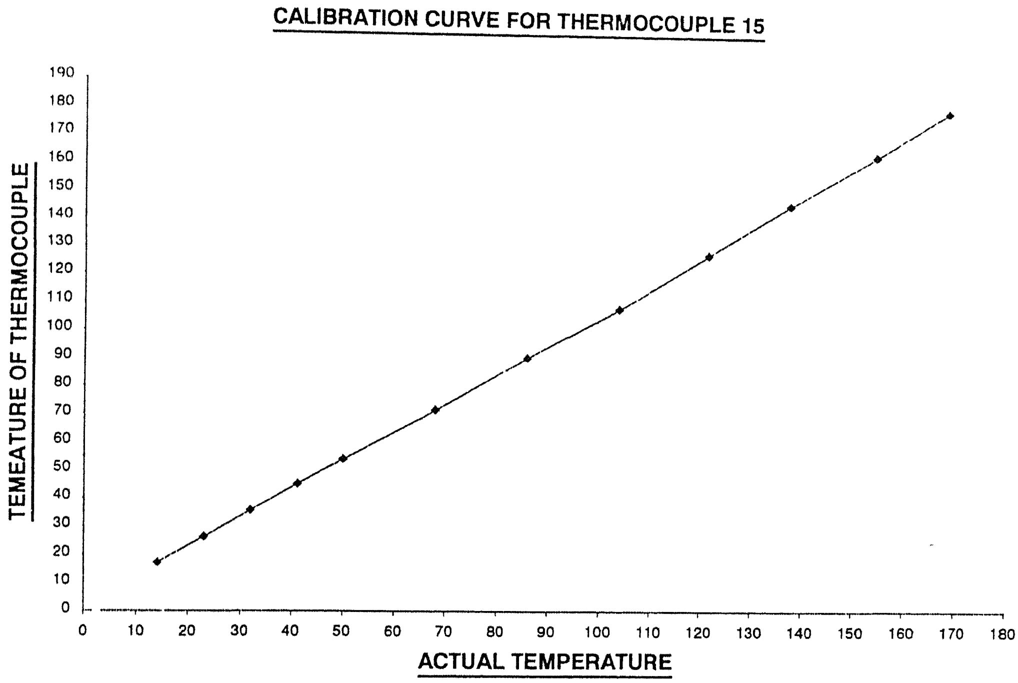


Figure 3.24 Calibration Curve for thermocouple'T15'

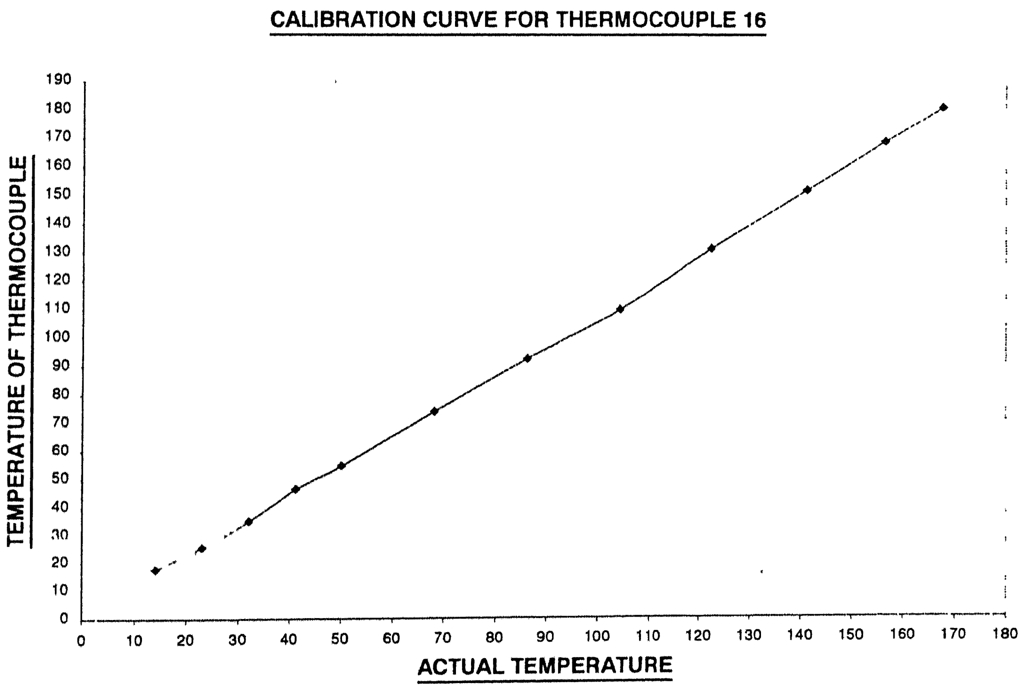


Figure 3.25 Calibration Curve for thermocouple'T16'

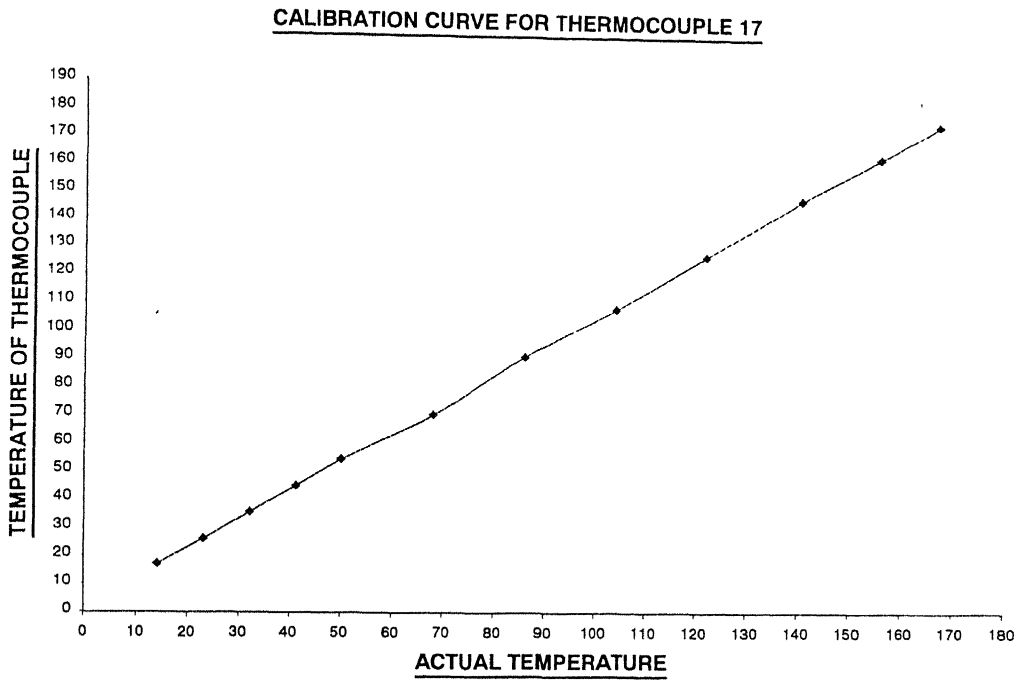


Figure 3.26 Calibration Curve for thermocouple'T17'

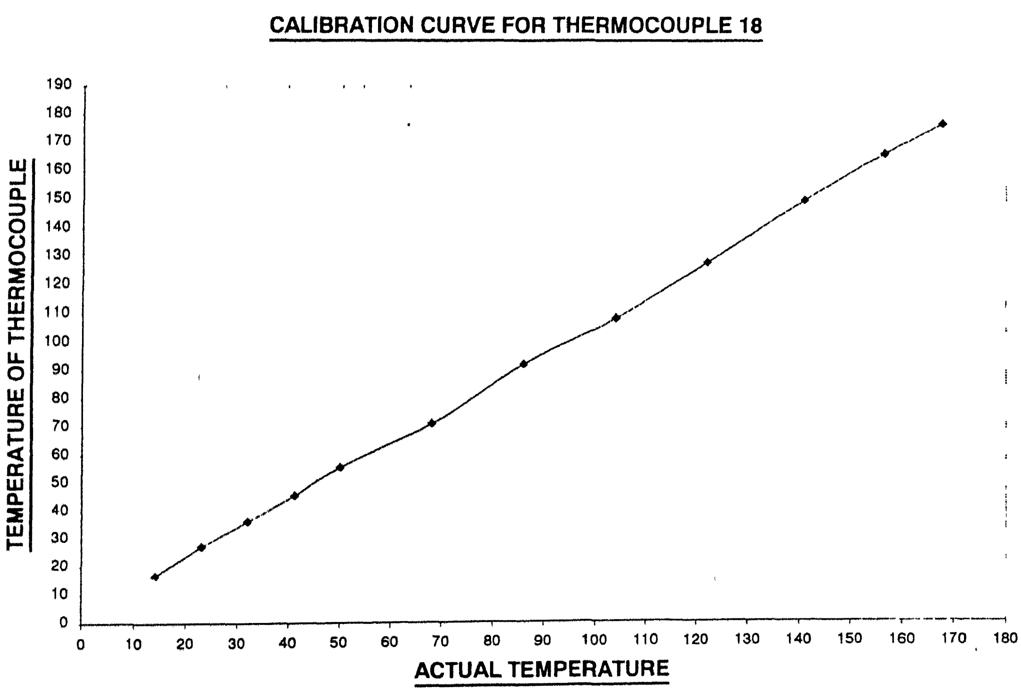


Figure 3.27 Calibration Curve for thermocouple'T18'

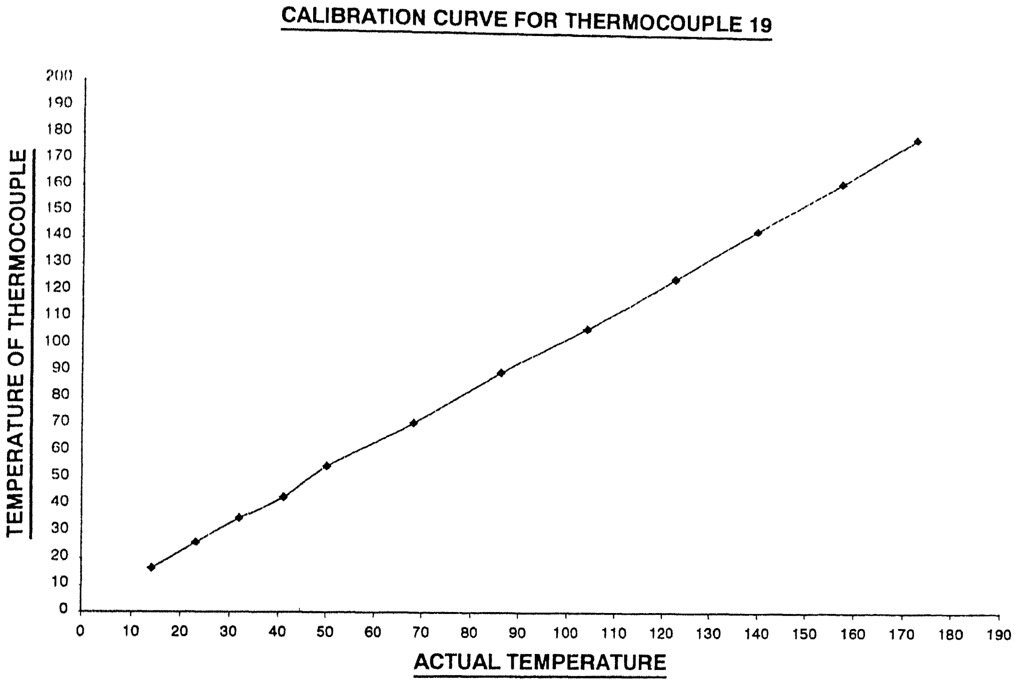


Figure 3.28 Calibration Curve for thermocouple'T19'

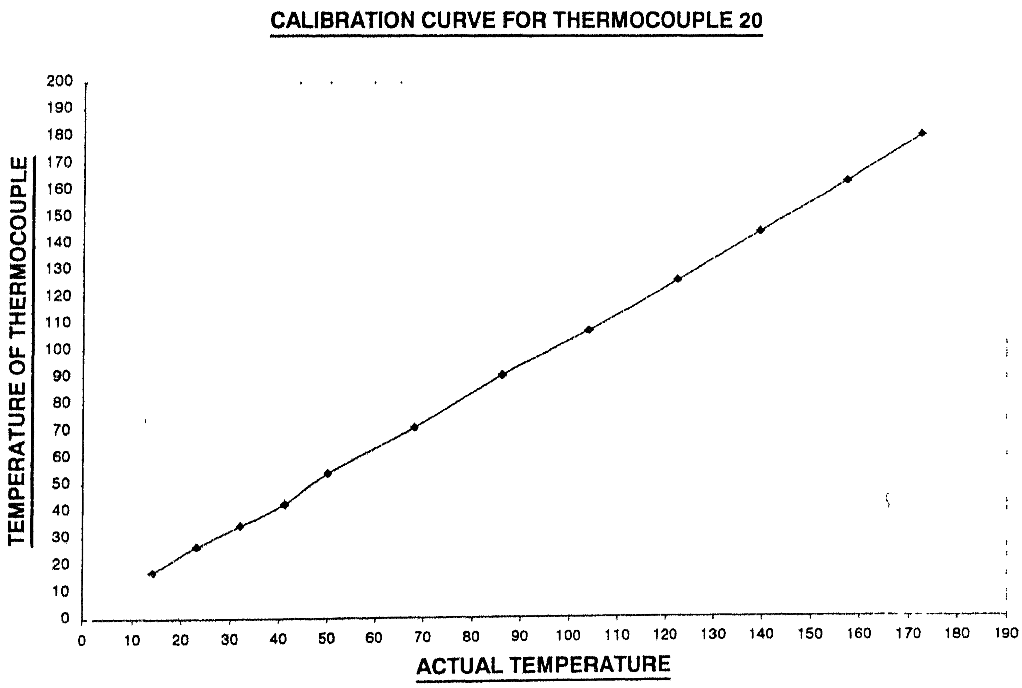


Figure 3.29 Calibration Curve for thermocouple'T20'

end of the 9.525 mm (3/8") copper 'T'. The free end of the 9.525 mm (3/8") diameter tube of 101.2 mm (4") length inserted in the 15.87 mm (5/8") diameter tube serves the purpose of a thermocouple well which can be filled with silicone oil and the thermocouple inserted in it to give the temperature at the exit of the condenser.

(ii) For the test sections. For measurement of temperature at various points on the test section, it is not desirable to insert the thermocouple bead inside the capillary tube, as it will almost block the small diameter bore. Since the beads are not being inserted inside the capillary tube, thus, only surface temp at those points where pressure tappings have been taken needs to be measured. This was done by soft soldering of thermocouple beads at the required points and connecting them to a central temperature measuring instrument as has been done for measuring temperature of six points on the main line.

(c) Identification of thermocouples. The thermocouples have been given a nomenclature as listed in table 3.6 for the purpose of identification as to which thermocouple represents temperature at which point.

(d) Connections to Temperature measuring instrument. The thermocouples attached in the main line are of a very high quality thermocouple wire and thus keeping the cost factor in mind, cannot be used as the lead wires to connect the thermocouples to the instrument. Thus, suitable lead wires are taken to connect these thermocouples to the various terminals marked 1 to 21 in serial of the temperature-measuring instrument. By switching on that particular terminal, the instrument shall indicate temperature corresponding to that point. The readings of temperature obtained from the thermocouples are prone to errors due to electrical noise from the surroundings e.g. fluorescent tubes, electric motor, etc, though for 24 gauge thermocouple wires it generally does not become a problem. However, to eliminate any chances of such errors creeping in the readings, the lead wires have been wrapped with aluminium foil and suitably earthed using crocodile clips.

Table 3.6. Nomenclature of Thermocouples

Sl. No	Location of Points	Nomenclature Of Thermocouple
1	**Test section** Starting from the drier side towards the evaporator- Point number 1 to 13	'T1' to 'T13'
2	**On the main line** (a) Exit of the test section or inlet of the evaporator, (b) Inlet of the compressor (c) Exit of the compressor (d) Inlet to the condenser (e) Exit of the condenser (f) Exit of the flow meter/Inlet of the expansion device	'T14" 'T15" 'T16" 'T17" 'T18" 'T19"
3	**Miscellaneous** (a) Inlet of the test section (b) Ambient	'T20" 'T21"

3.5.4 Power Supply

The power supply is taken from a 440 Volts, 50 cycles/sec, 3- phase mains, through a 3- phase voltage stabilizer of 400V, 50 Hz with capacity of 10 KVA which can take a maximum load of 14.6 Amps. The stabilized output is fed to the 3-phase compressor motor via an energy meter, a starter and an on-off switch as shown in figure 3.10. Separate connections are provided for the operation of the solenoid valves and the 1.5 HP motor of the evaporator. In addition 2 Nos of 15 amps, 3 pin sockets are provided on the panel board to enable fixing any temperature measuring instrument or other gadgetry that may be required later. The electrical circuitry of control panel switches and for the main supply is given in figures 3.30 and 3.31 respectively.

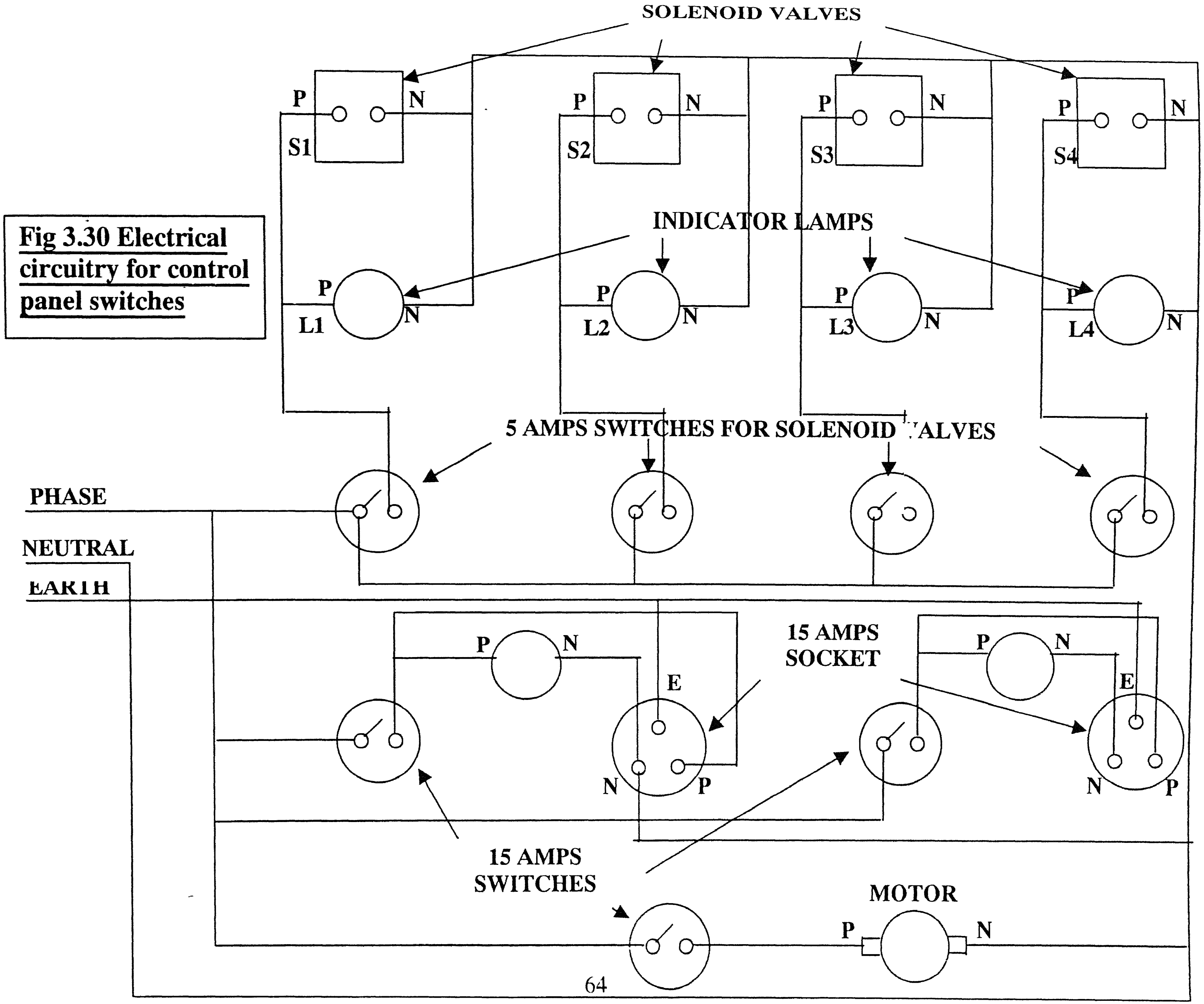


Fig 3.30 Electrical circuitry for control panel switches

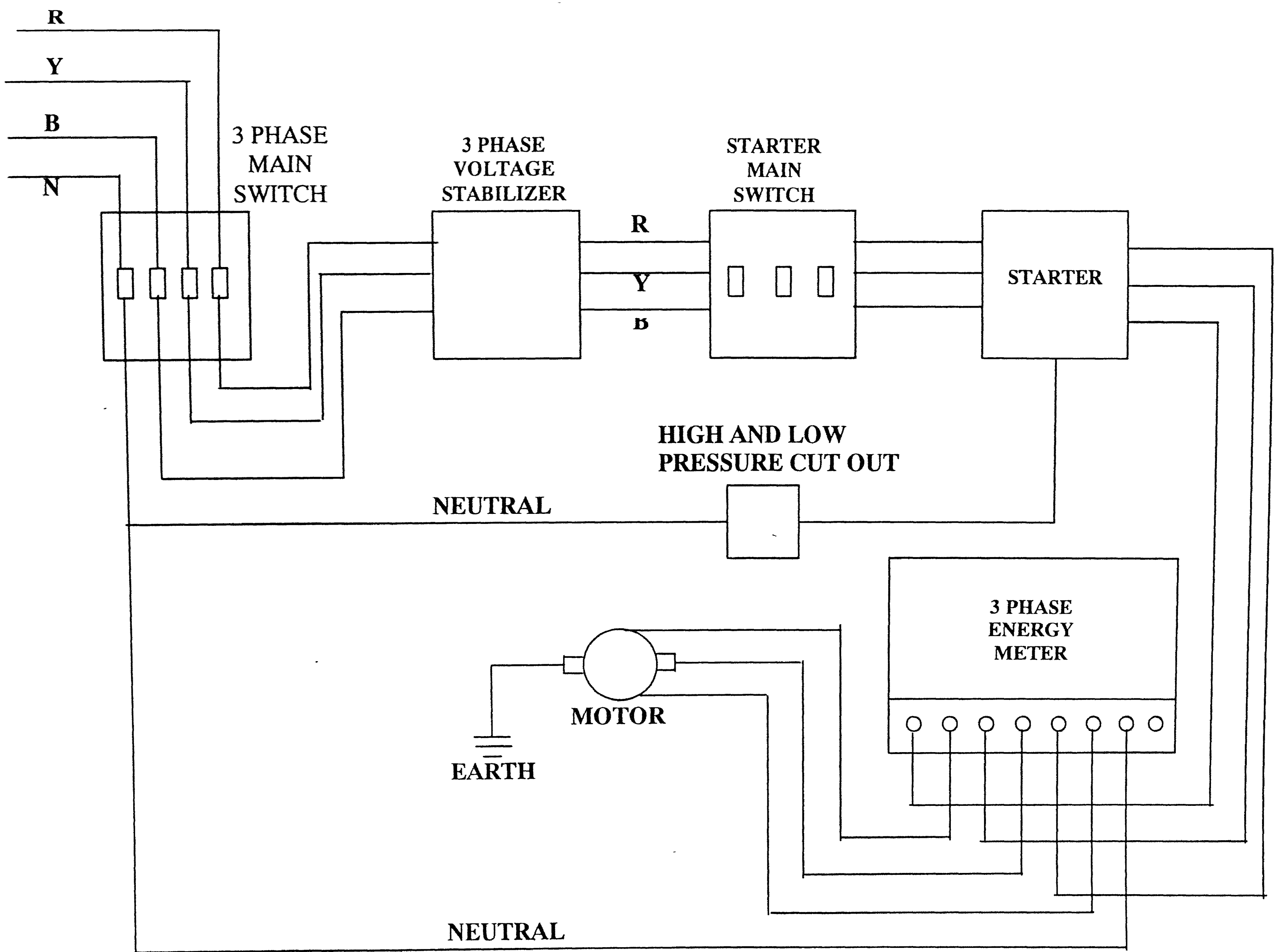


Figure 3.31. Electrical Circuitry of main supply to Motor

3.5.5 Energy measurement

A calibrated 3-phase energy meter is used to measure the input to the motor driving the two ton, open type of compressor. The specifications of the energy meter are as under.

(i) Type: 3 Phase, 4 wires.

(ii) Voltage: 240 V.

(iii) Amperage: 10-40 Amps.

(iv) Accuracy: Class 1 as per ISI 13779,

(v) Make: Larsen and Tubro.

(vi) Model: EM 301.

3.5.6 Water Flow Measurements

A Brook's rotameter of maximum flow measuring capacity of 10 Gallons per Minute is connected at the inlet of the condenser for the measurement of the flow rate of the cooling water. However, in the present study, the rotameter has been used only to control the cooling water flow rate. Normal tap water is supplied for cooling of the condenser from a reservoir through a 1.5 HP pump which is continuously fed with fresh water. The hot water returning from the condenser is discharged to the drain. The water flow rate is controlled by means of a 12.7 mm (½") diameter hand operated valve placed in the inlet as shown in figure 3.1.

3.5.7 Leak Proofing of the system

All the removable single components e.g. the flow meter, the condenser and the oil separator are individually made leakproof before incorporating them into the system. The flow meter, the condenser and the oil separator are tested for leaks under water at a pressure of 200 PSI, using high pressure Nitrogen. Any leaks noticed in the components are mended and the components put under 24-hour test of giving a steady indication of the respective gauge readings over the entire duration. The entire system including the test section is, then, checked for leaks at 200 PSI. Each test section, fitted with all the pressure tappings, is also checked for leaks at 200 PSI, under water before being installed in the system.

3.5.8 System Insulation

The suction line, the liquid line, and other connecting lines are all insulated with 6 mm diameter asbestos rope and then wrapped with insulation foam. The oil separator, flow meter, condenser and the oil separator are insulated with 75 mm thick glass wool and wrapped with insulation foam. The flow meter is further covered with polythene to provide extra insulation. All the lines in the system including pressure header connections are insulated with 6 mm diameter asbestos rope. In addition, the lines from the flowmeter outlet to the test section inlet, from the test section outlet to the evaporator inlet and the pressure header connections are all covered with glass wool, 50 mm thick.

3.5.9 High and Low pressure cut out

For the safety of the motor, it is essential to provide a low and a high pressure compound cut out in the system so that the compressor switches on only within the set values of high and low pressures. The cut out has an arrangement for fixing of the lower and upper limits of working pressure. Say, the lower limit of the evaporator pressure has been fixed at 20 PSI (1.38 bars) and the higher limit of the condenser pressure has been fixed at 150 PSI (10.35 bars). The compressor will start operating at 20 PSI (1.38 bars) pressure and will remain switched on till the time the pressure is below 150 PSI (10.35 bars). The moment the pressure in the delivery line goes beyond 150 PSI (10.35 bars), the compressor shall switch off. Thus to sense the operating pressures in the system, a 6.35 mm (¼") diameter copper tube with one of its ends flared and fitted with 6.35 mm (¼") brass nuts, is brazed at the copper 'T' at the pressure holes before and after the compressor. These two lengths are connected to the arrangement provided for in the cut out i.e. the point before compressor is connected to the point in the cut out meant for sensing low pressure limit and the point after compressor connected to the point in the cut out meant for sensing higher pressure limit.

3.6 Experimental Procedure

3.6.1 Fixing up of the test sections

Experiments are started using the longest lengths for each test section, which are subsequently reduced in stages to yield a number of test samples. Having done all the checking for any leakages, etc; the prepared test section with a starting length of 3.6 m is connected to the system. All the tapping lines of the test section are also connected to the respective Hoke's valves. Route 1 is cut off by closing valves DV5 and DV6. To check any leakages in the connections so made, nitrogen gas is once again passed in the system to a pressure of not less than 150 PSI. If there are no leaks detected in the connections of the test section, the system can be considered to be ready for charging. Leak testing of a particular test section is required to be done at least once, i.e. at the time of fixing the test section of the longest length, i.e. of 3.6 m. For example, straight test section of 3.6 m at the time of fixing in the system will have to be checked for the leakage. Later when the length of this test section is reduced care should be taken to close all the Hoke's valves V1 to V13 and also valves DV7, DV8, and DV9. The length of the test section can then be reduced as desired and in place of the cut portion of the test section, 12.67 mm (½") diameter copper tube should be brazed without opening the connections at any of the three valves DV7, DV8 and DV9.

3.6.2 Starting length of the Test Section

For each configuration, specimen of the largest length i.e., 3.6 m is taken first and studies are made. Subsequently the length is gradually reduced in steps of 15 cms to obtain test sections of same configuration of different lengths. In deciding the maximum starting length for the test sections, due note is taken of the capacity, of both the flow meter and the receiver. This is necessitated by the fact that the flow rate invariably decreases with the increase in the length of the capillary tube, of a particular diameter. This decrease in the flow rate creates a capacity balance problem which, in our case, would be characterized by a slow fall in the level of the refrigerant in one of the flow meter limbs and a sharp rise in the level of the refrigerant in the other. This would finally result in the flow of the refrigerant from one tube of the flow meter into the other. The maximum lengths of the test sections are, therefore, arrived at, after actual testing of the capillary sections which implies that initially the starting length of

each configuration can be kept as 3.6 m only, and if there is capacity balance problem then the length can be reduced gradually. The length at which the capacity balance problem is not encountered can be taken as the actual starting length for that configuration and diameter of the test section.

3.6.3 Charging of the system

Once the system is made 100% leak proof, the system can be charged with the desired refrigerant. Before charging, the system is flushed with Nitrogen gas to remove any traces of moisture and oxygen. The system is completely evacuated and then flushed with a small quantity of the refrigerant to be used for actual experimentation. For flushing purposes Route 1 is used. This is to ensure that the oil that may be present in the system is discharged to the surroundings and does not choke the capillary during the actual experimentation.

The system when undercharged showed a lot of frost formation on the evaporator coil and also the suction line of the compressor. When it was overcharged, the evaporator coil showed sweating on the outside surface. It was only when right amount of refrigerant was charged, the system operated at a rated capacity with minimal frosting.

3.6.4 Switching On the System

After mounting a particular test section into position, the system is switched on. Before taking down any measurements, the system is run till such time the desired inlet conditions at the test section are obtained. This invariably takes about half an hour duration. The desired inlet condition in terms of liquid saturation temperature can be obtained by means of adjusting the flow rate of the cooling water for the condenser. Since no provision has been made to study the effect of degree of subcooling, this is the only control available to us. Thus, as and when the inlet conditions at the test section are reached corresponding to the condenser temperature and pressure, the readings can be taken.

3.6.5 Data to be Recorded

In light of the effects to be studied, following readings are taken once the system reaches the desired inlet conditions of the test section

(a) Temperature and pressure at all the points of the main line and the test section are required to be measured. Temperature at the desired point is taken by pressing the respective switch of the 48-point temperature recorder. The thermocouple of that point would indicate the temperature at that instant.

(b) Pressure at any point can be found out by opening the respective Hoke's valve and by closing either the high or low pressure gauge of the respective header line depending upon the point for which the pressure is being measured. Care must be taken in closing the low pressure gauge valve if pressure is being measured for a point on the high-pressure side else there may be a chance of the low pressure gauge exploding due to high pressure. It is very evident for the points on the main line as regards their being on high or low pressure side, however it is not so for the points at the test section. Thus, while measuring pressures for the points at the test section, it is advisable to first open the high pressure gauge. If the reading shown by the gauge is smaller than the upper limit of the low pressure gauge then only the low pressure gauge should be opened to get exact reading of the pressure.

(c) The mass flow rate of the refrigerant can be determined by observing the decrease in the volume of the refrigerant in the flow meter limb over a noted period of time as explained in section 3.5.1(c).

The above readings along with the diameter, length and shape of the test section would constitute one set of data. In a similar manner, different sets of readings are to be obtained for various lengths, diameter and shape of the test sections and variations plotted on a graph to give the desired effects.

3.6.6 Discharging the system

The system is required to be discharged from time to time. The various occasions when the system is discharged are as follows:-

(a) Before changing the test sections,

(b) When any leakage is found after the system has already been charged,

(c) When a component of the system is required to be changed,

(d) On completion of the entire set of readings.

The procedure of discharging the system is as follows

The discharge line is connected to the empty or partially filled cylinder of the refrigerant being used, in this case R-134. The other end of the discharging line is connected to the discharge valve of the compressor. The compressor is now run till such time the entire refrigerant in the system is accumulated in the cylinder. Since, there is no refrigerant fed in the suction to the compressor, no refrigerant is thus recirculated in the system and after some time of running the system most of the gas is collected in the cylinder.

Chapter4

Preliminary Validation of the Experimental Data

4.1 Introduction

The system was tested after charging it with R-12 refrigerant to validate the functioning of various measuring arrangements that have been incorporated in the set-up. Data were obtained for a particular length of the capillary tube of straight configuration and for a particular condenser pressure. These data were taken basically to study the pressure and temperature drop in the capillary and to confirm its trend with the available data reported earlier for R-12 (42). These data obtained are represented graphically in Figures 4.1 and 4.2. However, a detailed data will have to be taken for R-12 for different condenser and evaporator pressures to confirm/validate the data reported in the literature earlier.

4.2 Parameters

The various parameters were as follows:-

(a) Date – 02 Jan 2002

(b) Place – Refrigeration and Air-conditioning Lab., Southern Labs., IIT, Kanpur.

(c) Total time taken by the system to come to steady state- 53 mins.

(d) Total duration of taking readings – 30 minutes

(e) Water cooling rate – 0.227 l/s (3.6 Gallons per minute).

(f) Ambient temperature – 17.23 °C .

(g) Energy meter reading –

At 12.15 PM- 24.10 kWh

At 12.45 PM- 24.90 kWh

(h) Test section details

Length – 3851 mm

Shape - Straight test section with two 'L' bends

Type - M 90

(i) Readings for finding mass flow rate of the refrigerant

l_1 = 370 mm =0.37m

l_2 = 251 mm =0.251m

$t_2 - t_1$ = 14.4 s

d = .075 m

4.3 **Pressure measurements**

(a) At test section pressure points

Table 4.1. Pressure measurements at various points on the test section

Sl No	Hoke's Valve No.	Capillary Length (mm)	Absolute Pressure (psi)	Absolute Pressure (bars)
1	21	At the inlet	133.5	9.207
2	1	150	126.5	8.724
3	2	450	125.5	8.655
4	3	750	120.5	8.310
5	4	1050	119.5	8.241
6	5	1350	117.5	8.103
7	6	1655	114.5	7.896
8	7	1970	109.5	7.552
9	8	2280	106.3	7.331
10	9	2575	101.9	7.027
11	10	2875	98.2	6.772
12	11	3155	90	6.207
13	12	3455	80.5	5.550
14	13	375.5	72	4.965
15	16	3851 (At the outlet)	56	3.862

(b) At main line pressure points

Table 4.2. Pressure measurements at various points on the main line

Sl No	Hoke's Valve No.	Pressure tapping location	Absolute Pressure (psi)	Absolute Pressure (bars)
1	16	Inlet of evaporator	56	3.862
2	17	Exit of evaporator	56	3.862
3	18	Discharge of compressor	136	9.379
4	19	Inlet to condenser	135	9.310
5	20	Exit of condenser	135	9.310
6	21	Inlet to test section	133.5	9.207

4.4 Temperature measurements

(a) At the test section points

Table 4.3. Temperature measurements at various points on the test section

Sl No	Thermo-couple No.	Capillary Length (mm)	Measured Temperature (°F)	Actual Temperature (Calibrated) (°F)	Actual Temperature (°C)
1	19	Start point	76	79	26.11
2	1	150	75	78.9	26.05
3	2	450	75	78.9	26.05
4	3	750	75	78.7	25.95
5	5	1350	75	78.9	26.05
6	6	1655	74	78.22	25.67
7	7	1970	71.5	73.3	22.94
8	9	2575	70	71.8	22.11
9	10	2875	69.5	69.8	21
10	11	3155	68	68.3	20.16
11	12	3455	66	66.0	18.89
12	13	3755	58.5	58.5	14.72
13	14	3851	43	44.6	7

(b) At various points on the main line

Table 4.4. Temperature measurements at various points on the main line

Sl No	Thermo-couple No.	Location of The Thermocouple	Measured Temperature (°F)	Actual Temperature (Calibrated) (°F)	Actual Temperature (°C)
1	14	Inlet of evaporator	43	44.6	7
2	15	Exit of evaporator	43.5	44.6	7
3	16	Discharge of compressor	176.5	178	81.11
4	18	Exit of condenser	80	82	27.78
5	19	Inlet to test section	77	79	26.11

4.5 Analysis of the system

(a) Refrigeration cycle on p-h diagram

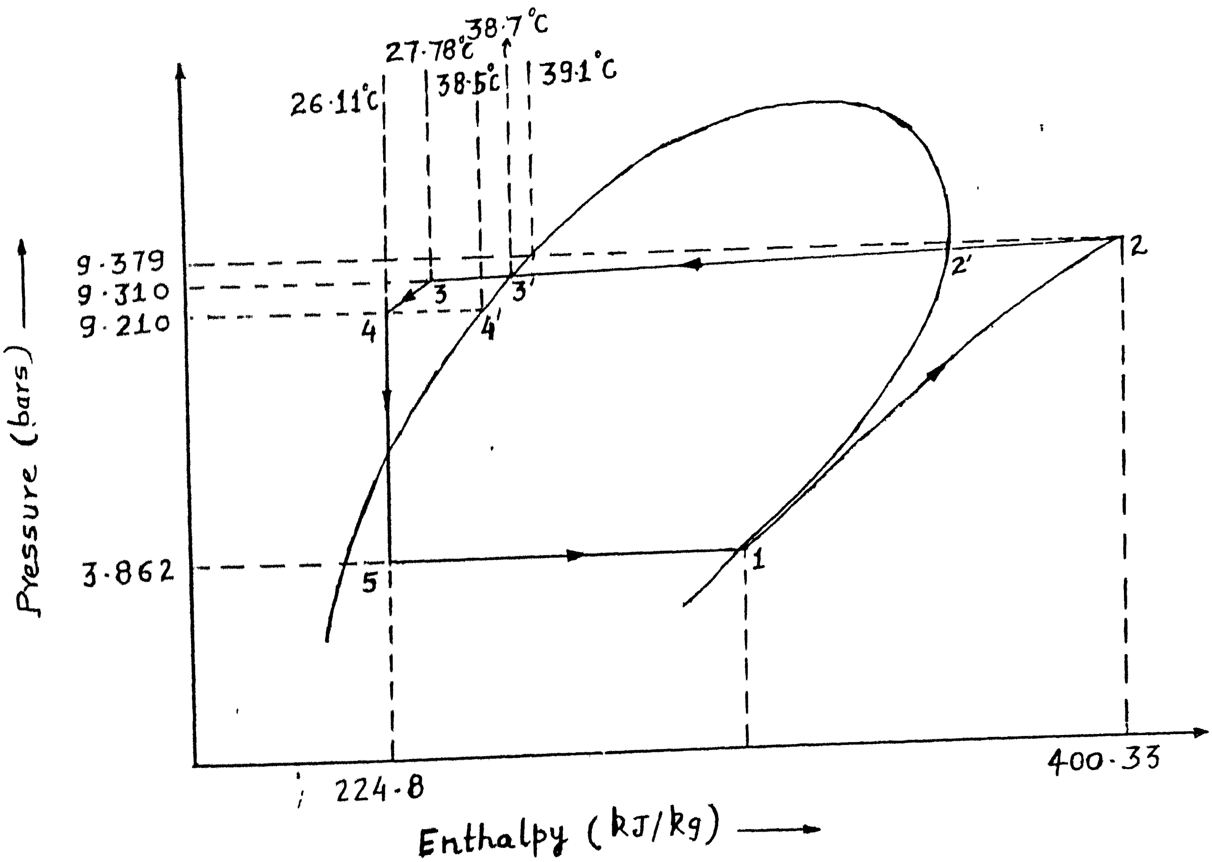


Figure 4.1 Refrigeration cycle on p-h diagram

(b) Enthalpy of the refrigerant vapor at the discharge of the compressor

Enthalpy at state point 2 $= h_2 = h_{2'} + c_{p_{2'}}(T_2 - T_{2'})$

$h_{2'}$ at discharge pressure of 9.379 bar (from saturation table for R-12)= 368.41 kJ/kg

Also, c_{p_v} = 0.758 kJ/kg k

$T_2 = 81.11^oC$

$T_{2'} = 39^oC$

Therefore, h_2 =368.41 + 0.758 (81.11-39) =400.329 kJ/kg

(c) Enthalpy of the refrigerant at the inlet of the test section

At 9.30 bar, saturation temperature of the refrigerant is 38.5°C and

$h_{4'}$=237.5 kJ/kg.

Therefore,

$h_4 = h_{4'} - c_{pL}(T_{4'} - T_4)$

$= 237.5 - 1.025\ (38.5 - 26.11) = 224.8$ kJ/kg

Assuming an isoenthalpic process from state point 4 to 5, enthalpy of the refrigerant at the end of the test section, h_5 = 224.8 kJ/kg

(d) Refrigeration effect

Since, no superheating of the refrigerant has taken place in the evaporator, thus the refrigerant entering the compressor can be assumed to be at saturated vapor state.

Enthalpy of the refrigerant at state point 1 (at evaporator pressure of 3.862 bar).

h_1 = 355.75 kJ/kg (from the refrigerant table for R-12)

Therefore,

Refrigeration effect, RE = $h_1 - h_5$

$= 355.75 - 224.8 = 130.95$ kJ/kg

(e) Mass flow rate of the refrigerant

(i) Mass flow rate for the rated 2TR refrigeration system

$2 \times 3.5167 = \dot{m} \times RE$

$$\Rightarrow \dot{m} = \frac{2 \times 3.5167}{130.94975} = 0.0537 \text{ kg/s} = 53.71 \text{g/s}$$

(ii) Actual mass flow rate of the refrigerant.

It was measured from the change in the level of the condensate in the flow meter and is given by the relation:-

$m = V \times \rho_l$, where,

V = Volume flow rate of the refrigerant $= \dfrac{\pi \times (d^2) \times (l_1 - l_2)}{4 \times (t_2 - t_1)}$

ρ_l = Density of the liquid at the outlet of the condenser=1258 kg/m^3 (from the saturation table for R-12)

$$V = \frac{\pi(.075^2)(0.37 - 0.251)}{4 \times 14.4} = 36.49 \times 10^{-6}\ \text{m}^3/\text{s}$$

Therefore,

m =3.6490234 x 10^{-5} x 1258 = 0.0459047 kg/s = 45.90 g/s

The mass flow rate of the refrigerant obtained theoretically for the rated capacity of the system is slightly different from the one obtained experimentally which is 0.04590 kg/s. The error may be attributed to faulty reading of levels from the flow meter.

(f) COP of the system

$$\text{COP} = \frac{m \times RE}{TotalWorkDone}$$

Power consumed by the electric motor in 30 mins duration =24.9 - 24.1

= 0.8 kWh

Power consumed by the 0.5 HP motor of evaporator and other electrical accessories used in the system (approximate) = 0.3 kWh

Therefore, total power consumed = 1.1 kWh

$$\text{Total work done} = \frac{1.1 \times 3600}{30 \times 60}$$

=2.2 kW

Therefore,

$$\text{Actual COP of the system} = \frac{0.459047 \times 130.94975}{2.2} = 2.73$$

4.6 Pressure Distribution along the Length of the Capillary Tube

Figure 4.2 shows a graph between the pressure at various points on the test section vs the length of the capillary tube. This gives the same trend as reported in the literature earlier (42).

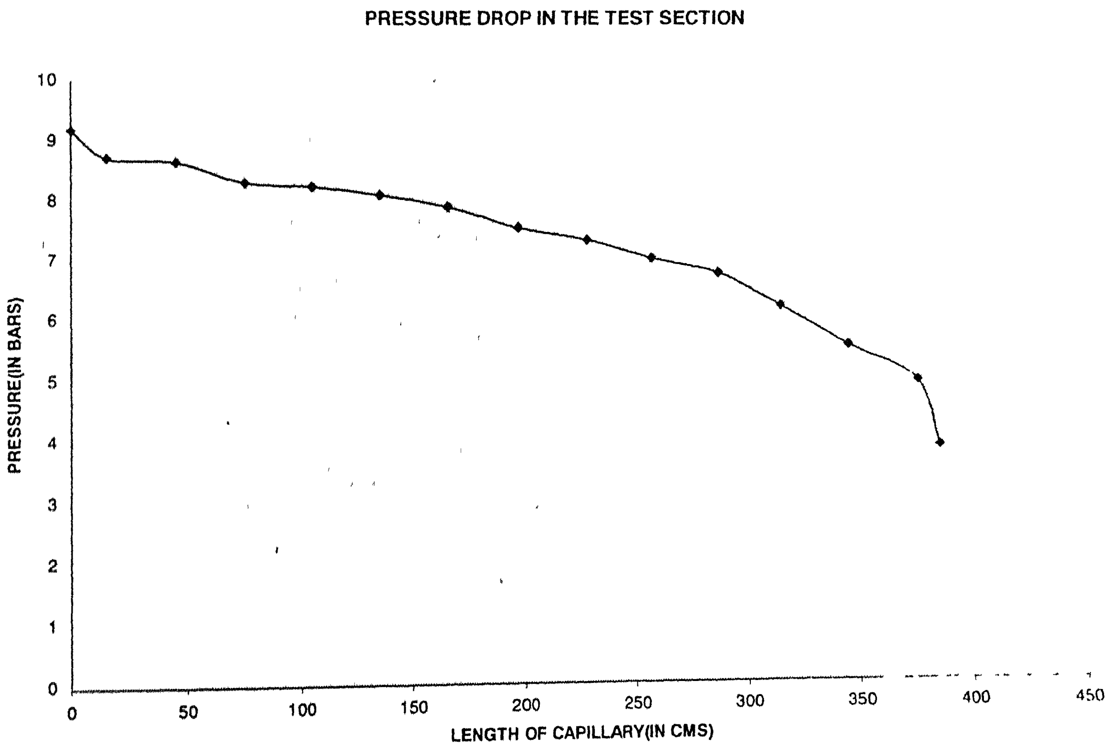


Figure 4.2 Variation in pressure drop along the length of the capillary tube

4.7 Temperature Drop along the Length of the Capillary Tube

Figure 4.3 shows a graph between the temperature at various points on the test section vs the length of the capillary test section. The trend of temperature drop along the capillary tube compares well with the data reported earlier in the literature (42).

4.8 Results and Discussions

(a) <u>COP.</u> COP of the system was found to be reasonably good. Taking into account all the approximations and the errors that might have affected the data

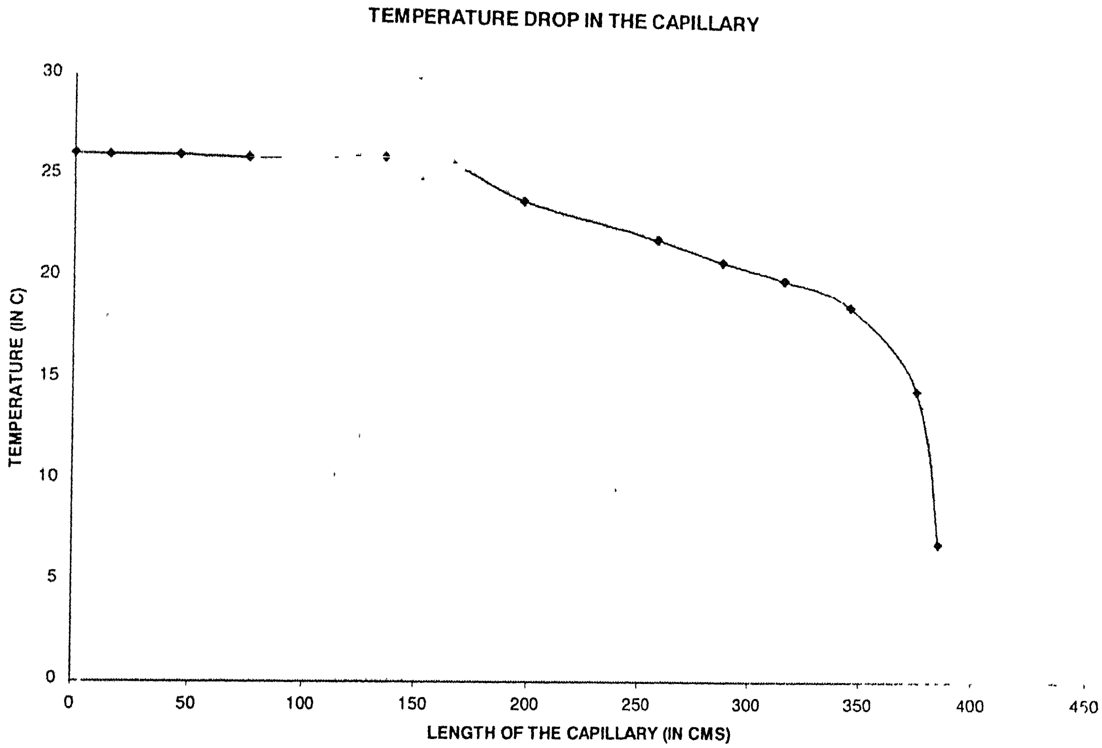


Figure 4.3 Variation in temperature drop along the length of the capillary

obtained, COP of the system can safely be taken as 2.6 which is absolutely reasonable indicating the good performance of the system.

(b) Mass flow rate of the refrigerant. As said earlier, there is a slight amount of variation in the mass flow rate of the refrigerant obtained from the set-up as against the theoretical calculations. This variation can be attributed to the error in noting the levels of the refrigerant in the flow meter.

(c) Plot for temperature drop in the capillary test section. The plot obtained for the temperature drop was found to be in agreement with the curves obtained by Giri (42). The temperature of the refrigerant in the capillary remained unchanged till a particular length from the inlet. It dropped gradually thereafter and after a particular point the temperature drop was found to be steep as shown in figure 4.3 above.

(d) Plot for pressure drop in the capillary test section. Figure 4.2 above shows that the pressure in the test section dropped gradually in the initial portion due to single phase flow and after the flash point was reached the pressure drop also increased due to two phase flow. This trend resembled with the results obtained earlier (42).

4.9 Reproducibility of the data

The system was made to run under almost the same ambient conditions the next day to confirm reproducibility of the data. The various operating parameters were kept almost the same and it was found that the data obtained the previous day was exactly reproducible confirming that the system was working absolutely fine except for a few refinements that may have to be undertaken at a later stage as suggested in the following chapter.

4.10 Conclusions

The results so obtained verify the functioning of the system. However, certain modifications and refinements have to be carried out before taking any additional data and compare them with the theoretical and experimental results of previous researchers. Since the data obtained from the set-up with R-12 compares well with the data of other workers, the test rig can be considered to be proper and the data as validated.

Chapter 5

Future Scope

5.1 Effects to be studied

Following effects are required to be studied to generate a data bank for different refrigerants. The starting refrigerant can be R-12 and further graduated to R-134, CARE and other desired refrigerants. R-12 is recommended to be the starting refrigerant since certain amount of data are available with R-12. This would enable validating the system and incorporating changes that may be required as an after thought. The effects to be studied are as follows:

(a) Effect of capillary length of a particular diameter and configuration i.e. straight, helical or spiral on the mass flow rate of the refrigerant, which in turn would effect the amount of refrigeration obtained at the evaporator.

(b) Effect of capillary diameter of each configuration on the mass flow rate of the refrigerant used.

(c) Effect of sub cooling on the mass flow rate of the refrigerant for each diameter of a particular length and for each configuration.

(d) Comparison of mass flow rate for various lengths of different configurations with the theoretical predictions.

(e) Comparison of the pressure drops in the capillary tubes of various configurations and their comparison with the theoretical models proposed and to be developed later.

5.2 Test Sections

For demonstration purposes, a single straight test section with all its connections has been prepared and fitted in the experimental setup. However, to get a reasonable data base for the capillaries, for different refrigerants, a number of test sections are required to be prepared which are as listed in following paragraphs.

5.2.1 Test Sections. A significant data base can be created by studying the effect on the mass flow rate of the refrigerant for three different diameters of the capillary. Ideally speaking, at least 5 to 6 different diameters of the capillary should be studied. However, a minimum of following numbers of test specimens of different geometry are to be fabricated for each diameter capillary:-

- One of straight configuration,
- One of helical configuration of D/d ratio = 15, and
- One of spiral configuration.

Thus for each diameter of the capillary, three test sections are made of 3.6 m length each, overall a total of 9 test sections.

5.2.2 Straight Test Section. The straight test sections are the simplest to make and assemble in the set up as has been done in the existing setup. Since the length of 3.6 m is big enough to fit within the space available between valves DV8and DV9, the test section can be fixed as demonstrated in the setup. The connection to the pressure tapping can be done as explained in Section 3.5.2 (c)(iii).

5.2.3 Helical Test Section. The second configuration of the test section is of helical shape. The test section capillary can be given a helical shape by winding the capillary around a wooden mandrill. The wooden mandrill should not be made of an arbitrary diameter for it shall cause problems if the experiment is required to be repeated with this mandrill having been lost. It is desirable to maintain a particular D/d ratio while deciding on the diameter of the mandrill, where,

D = Diameter of the mandrill + diameter of the capillary (d), and,

'd' is the diameter of the capillary.

The schematic of the proposed helical test section mounted on a wooden mandrill is given at Figure 5.1.

(a) Finding diameter of the Mandrill. To find the diameter of the mandrill the first step is to find the outer diameter of the capillary ('d') accurately with the help of an accurate micrometer or a vernier calliper. Assume D/d ratio of the helical test section to be as 15. Thus, by knowing'd', value of 'D' can be determined.

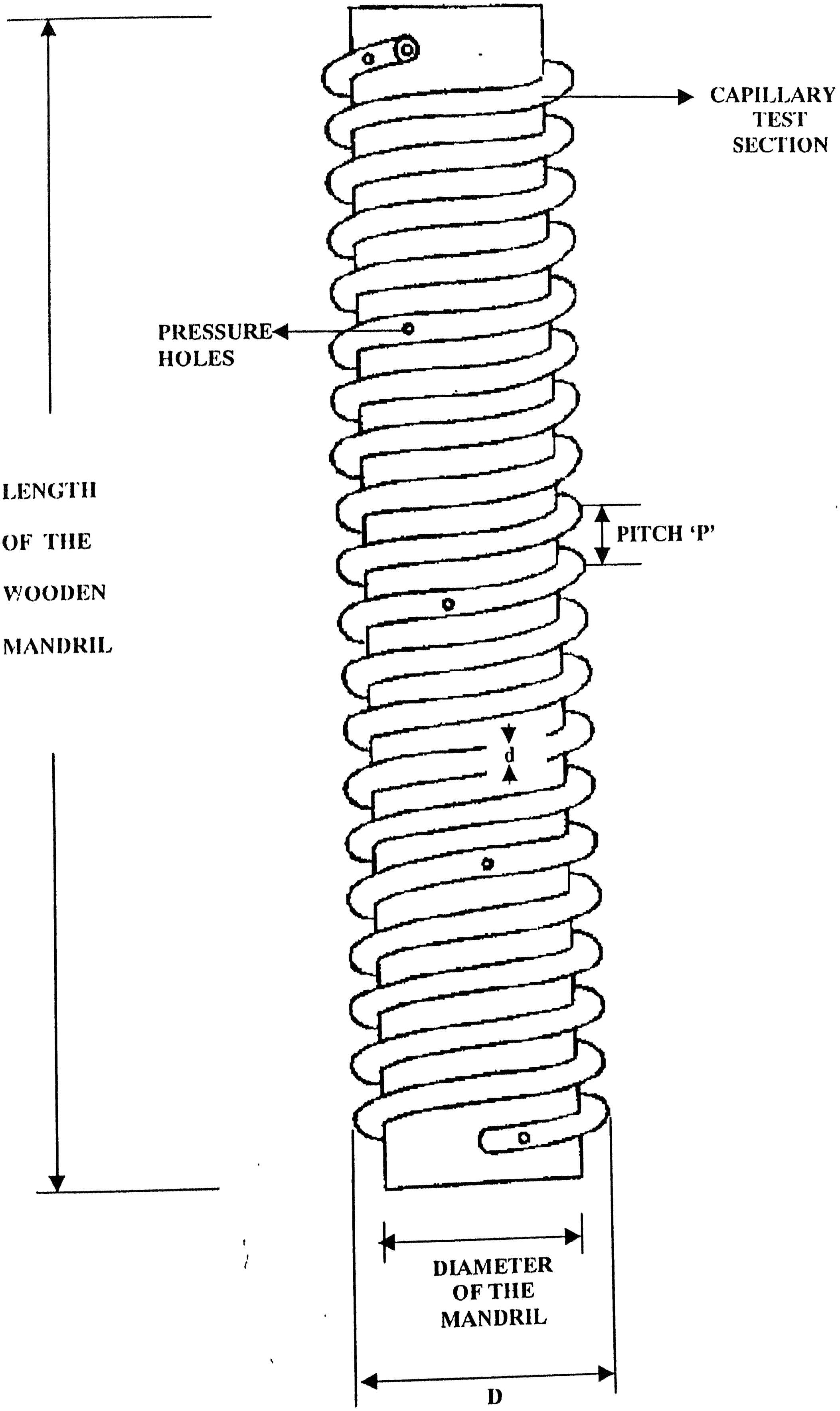


Figure 5.1 Schematic of proposed helical test section

Therefore, Diameter of the mandrill= D-d = 14d (for the assumed D/d ratio)

(b) Fixing up of the capillary test section on the mandrill. To wind the capillary test section on the mandrill which should not change its position while carrying out the experiments, it is essential to make a sort of groove on the outer surface of the mandrill. The depth of this groove should be half the diameter of the capillary test section so that the O.D of the mandrill with capillary wound around it will be = Diameter of the mandrill + d. The width of the groove should be = d + 0.5 mm.

(c) Length of the wooden mandrill. Let us take a specific helix angle, θ =4.12 degrees. Therefore, pitch can be calculated by the relation,

Tan θ = P/D, where, P = Pitch

=> P=D tan θ

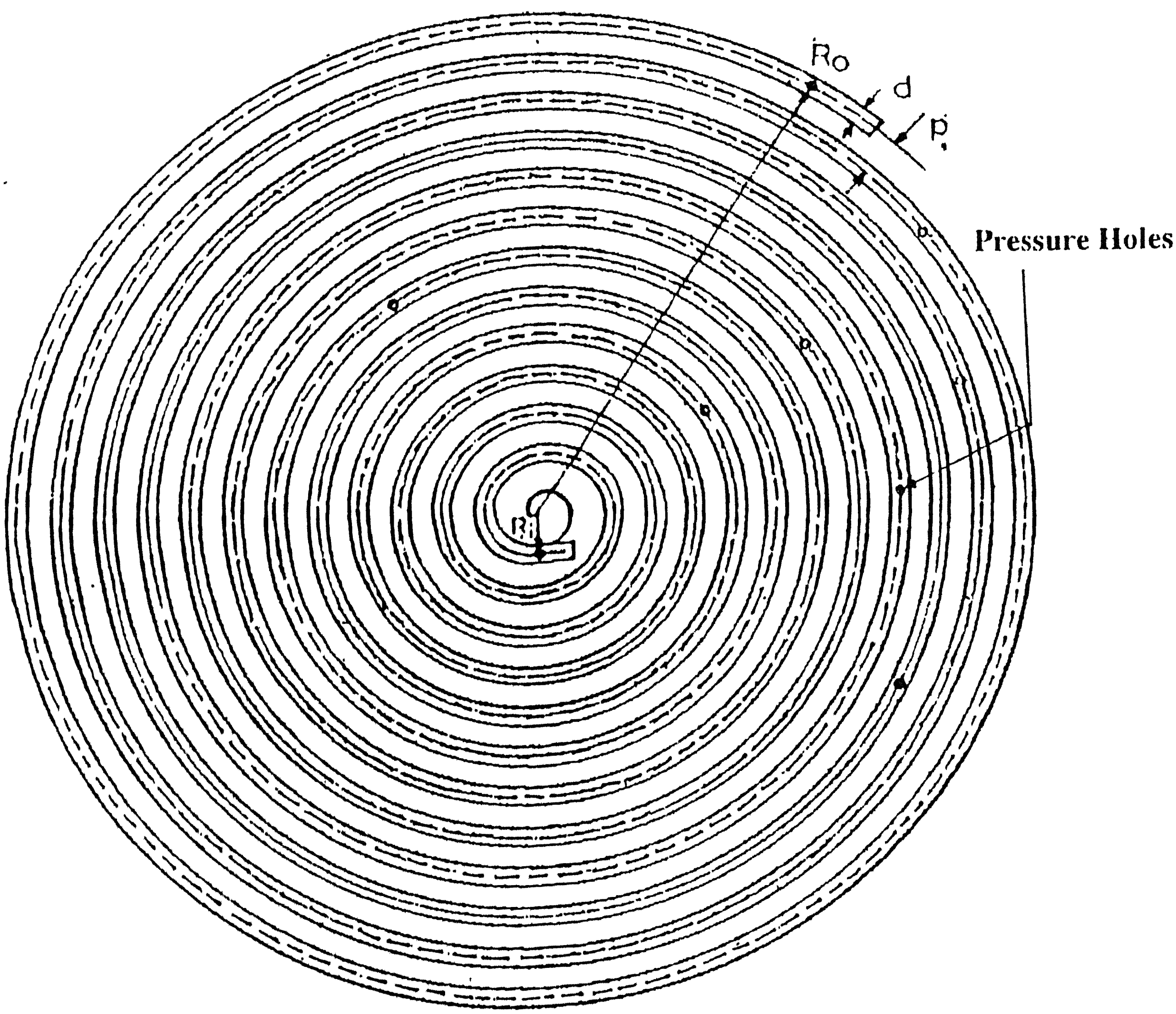
Since the wooden mandrill has to cater for the largest length of the test section, which in this case is 3.6m, therefore,

N π D = 3.6m, where, N= Number of turns.

Therefore, Length of the wooden Mandrill, L = N x P

5.2.4 Spiral Test Section. For making of spiral test section a 12.67 mm (1/2") thick wooden block of size 0.25 m x 0.25 m is taken. Inner radius from where the coil is to be wound is to be kept at 5 mm and the pitch can be selected as 5 mm. The grooves can be cut circularly on this wooden block till an outer radius of 200 mm and of width 5 mm greater than the diameter of the capillary i.e, 'd'. The depth of the groove can be half the'd' so that the capillary doesn't slip from its position while carrying out the experiments.

It may be noted that the above values are not sacrosanct, but are the only recommended figures. Infact, the D/d ratios can be altered to find the effect of varying D/d ratios on the mass flow rate of the refrigerant. Similarly, all other parameters can be changed to find the effects. Schematic of the proposed spiral test section mounted on a wooden mandrill is given at Figure 5.2.



Where,

Ro = Outer Radius of the spiral shaped capillary test section = 20 Cms.

Ri = Inner radius of the spiral shaped test section = .005 m.

P = Pitch of the spiral test section = .005m.

D = Outer diameter of the capillary test section

Figure 5.2 Diagram of the proposed spiral test section

5.3 Possible Refinements in the Test Rig

The system could be further refined as far its evaporator design and connections are concerned. The various areas where refinement of the system could be considered are as follows.

5.3.1 Evaporator design. In the present system, the evaporator has been designed arbitrarily in the sense that since evaporator performance is not being studied at all, hence it is just catering for certain heat inputs from the surroundings which would eventually cause complete vaporization of the refrigerant. This heat gain by the refrigerant cannot be calculated in any way. Also, the evaporator has been slightly over designed to cater for the superheating of the refrigerant so that no liquid refrigerant is left that may enter in the compressor through the suction line. This arrangement may suffice during the summer season when the ambient temperatures are high, but during the winters the same system may not be sufficient to cause complete vaporization of the refrigerant due to lower surrounding temperatures. Thus, this system can be replaced by an evaporator which is designed to give a specific quantity of heat input to the refrigerant passing through it. The specific quantity of heat can be provided by means of heating tape which could be wound over the evaporator pipes and well insulated from the outer side so that entire heat generated in the heating tape is transferred to the refrigerant passing through the evaporator coils and the loss to the surroundings is minimized. An additional length of the superheating coils could be designed for increasing the refrigeration effect.

5.3.2 Liquid Separator. A liquid separator could be provided after the evaporator which could remove any traces of liquid refrigerant that might have escaped vaporization in the evaporator coils. However, liquid separators are generally not used for such a small capacity refrigeration system though there is no harm in using one.

5.3.3 Connections to the Test Section Pressure Tapping. In the present system, only one test section has been prepared with all its connections. The various pressure points of the test section are connected to the respective Hoke's valves by means of a small length of 6.35 mm (¼") copper tube which is brazed to the pressure holes and then by a

small length of 9.525 mm (3/8") diameter tube and subsequently with a connecting length of 6.35 mm (¼") copper tube till the 6.35 mm (¼") brass union of that particular Hoke's valve. The connection is further extended till the Hoke's valve by means of another 6.35 mm (¼") copper tube. This system is slightly cumbersome in terms of brazing work etc and also involves greater chances of leakage due to more number of connections. Also, this involves larger requirement of the refrigerant to fill in the pressure tapping before any pressure measurement at that point could be made.

This method can be refined if a smaller diameter capillary (smaller than the one used as expansion device) could be brazed at the pressure holes. This smaller diameter capillary can be extended till 152.4 mm (6") short of the respective Hoke's valve and can be brazed to 165.1 mm (6.5") long, 6.35 mm (¼") diameter copper tube which is flared at one end and fitted with a 6.35 mm (¼") brass nut. This brass nut can be tightened on to the Hoke's valve.

This method would not only reduce the number of joints i.e., at the union and the amount of brazing work, but also would require comparatively lesser amount of the refrigerant. However this method shall involve higher accuracy and greater degree of difficulty in brazing at the pressure holes.

5.3.4 Refrigerant Rotameter. In the present system, the mass flow rate of the refrigerant is measured by means of a flow meter. The flow meter is basically used since the effects to be studied are not only for a single refrigerant, but for a number of refrigerants. The flow meter thus can be used for all the refrigerants. However, flow meter has its own inherent disadvantages of not being precise and also errors involved while reading the levels. Use of rotameters for different refrigerants would obviate this problem though the cost factor may have to be considered before going in for this arrangement.

5.4 Shortcomings of the Present System

The system is not yet complete to finally go in for the readings directly. There are few short comings which are required to be attended to before any meaningful results could be taken. These drawbacks are as listed below.

5.4.1 Leak Proofing of the System. The system is not yet 100% leak proof though the leakages have been brought down considerably i.e. 0.5 PSI at 132.5 PSI in 7 hours duration. While carrying out the leak testing of the system it was noticed that the compressor might be leaking though the leakage is very minor. Before charging the system with actual refrigerant it is imperative that the system is made absolutely leak proof else the costly refrigerant will leak to the surroundings not only causing financial loss but also the environmental related problems.

5.4.2 Degree of Subcooling. No provision has yet been made in the set-up to provide any subcooling of the liquid refrigerant after the condenser. Subcooling is essential from the point of increasing the refrigeration effect. Also, for studying the effect of degree of subcooling, on the mass flow rate of the refrigerant, it is necessary to provide the arrangement for it. This can be done by passing the line after the flow meter through the evaporator of another refrigeration system designed for maintaining a specific temperature in its evaporator. The point to be kept in mind is that the surface area of the pipe passing through the evaporator of this secondary refrigeration system should be calculated precisely to enable the attainment of desired temperature drop in the liquid at the inlet of capillary thus obtaining different degrees of subcooling. This will require an additional refrigeration system. Alternatively, additional coils of different surface area can be connected condenser of the system using valves. This arrangement can provide different values of the degree of subcooling at the inlet of the capillary tube, without using a separate refrigeration system.

5.5 Sources of Error and Uncertainties. Various sources of error that might occur in the data during the experimentation over which the author has no control are listed below.

(a) The pressure drops occurring due to sudden change in diameter of the pipe lines connecting different points. Extreme care has been taken in making such connections, but they can be further reduced by minimizing sudden changes in the cross- sectional area of the connecting lines.

(b) The capillaries that shall be used in the experimentation should ideally have the same surface finish and diameter throughout its length. However, in practice it

may not be feasible to get such lengths in the open market and thus procurement of such test sections shall be a major problem in case very precise data are required to be obtained.

(c) The diameter of the test section would change at places where holes will be drilled to fix the pressure tapping. This is absolutely unavoidable if pressure measurements are to be done along the length of the capillary tube. The only way of reducing this error is to drill the holes and to clean the burrs very carefully. Also, while brazing the lines for pressure tapping on to these pressure holes extreme care will have to be taken so that the molten copper doesn't find its way inside the capillary thereby, either choking it completely or changing its diameter.

(d) The temperature and pressure measurements in the system may get effected due to reasons beyond the control of the person conducting experiments. The instrumental errors over a period of time may be a source of error.

5.6 Conclusion

(a) The present work aims at design and development of an experimental test rig for creating a data base for capillaries of different geometric and configurations for different refrigerants using a 2- ton vapor compression refrigeration system. All out efforts were made to make the system robust to withstand handling while conducting experiments over a considerable period of time. The system is complete in all respects except the additional refinements which may be implemented as a part of the future work.

(b)The leakages which are the inherent problems of any refrigeration system and also the most undesirable thing to happen had been the sour point of this thesis work. Inspite of all the efforts put in by the author, the leakages in the system couldn't be stopped altogether though brought down considerably to about 0.5 PSI pressure drop at 132.5 PSI in 7 hours duration. This would be the major task of any researcher who would be continuing on this project subsequently.

(c) The system was also tested for a particular condenser and an evaporator pressure and the results so obtained for the pressure and temperature drop in the capillary test section compared well with the data reported earlier in the literature. However, this data was obtained only to check the workability of the system. A detailed

data base for any refrigerant can only be obtained after incorporating the refinements as suggested in chapter5.

5.7 Suggestions for Future Work. Since retrofitting of small refrigeration systems with new refrigerants is almost a dire necessity now, it is extremely important to take up the following almost on an urgent basis.

(a) Modeling of flow through a capillary tube and comparison of its predictions with the real data with R-12 to validate the experimental data itself.

(b) Collect data with new refrigerants including hydrocarbons and their mixtures with various configurations of capillary tubes.

(c) Modeling of flow through capillary tubes of different geometric configurations and comparison of the predictions with the real data.

(d) Retrofitting a R-12 vapor compression system with new refrigerants and studying their performance and make meaningful recommendations.

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